Copyright, 1935, by the AMERICAN SOCIETY OF NAVAL ENGINEERS.

JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. 47. FEBRUARY 1935. No. 1

The Society as a body is not responsible for statements made by individual members

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published).

Admiral WILLIAM H. STANDLEY, U.S.N.
Captain N. H. WRIGHT, U.S.N.
Captain B. H. BRUCE, U.S.N.
Commander S. S. KENNEDY, U.S.N.
Captain HENRY WILLIAMS, (C.C.) U.S.N.
Commander H. N. PERHAM, U.S.C.G.
Mr. JOHN F. NICHOLS
Mr. W. M. CORSE
Commander C. S. GILLETTE, U.S.N.

METHOD OF TESTING HIGH PRESSURE AIR COMPRESSORS.

By R. L. DENNISON, LIEUT., U. S. N., MEMBER.

A survey of pertinent test codes, text books, and reports of air compressor tests is likely to leave one more confused than enlightened as to a proper method for testing high pressure air compressors. It will appear that not only is there a lack of uniformity in the manner of testing, but also considerable disagreement or lack of specific instruction concerning the analysis of the observed data to obtain quantitative results.

It is intended in this paper to give a brief discussion of the problem of testing high pressure air compressors. Several alternative test methods will be described. Correction factors will be discussed, and their use demonstrated by application to actual per-

formance computations. The test procedure to be described is based on practice at the U. S. Naval Engineering Experiment Station.

The basic test problem is to discover the relationship between the power required to drive the compressor and the amount of compressed air delivered. The determination of the first factor is relatively simple. The compressor may be driven by a dynamometer and the power computed from indicated speed and torque. In many cases the compressor and its drive are treated as a unit, and the power input to the driver only is measured. For turbine driven compressors the amount of steam, in pounds per hour with specified steam conditions at the throttle, is usually required and may be obtained by weighing the turbine condensate. In dealing with an electrical drive the input is easily determined from appropriate electrical measurements. The determination of the power required is quite direct, and should present no difficulty. On the contrary, the determination of the compressor output, while not particularly complex, presents many opportunities for error.

The output, or rating, of a compressor is usually stated as a certain number of cubic feet per hour. Does this apply to intake or discharge air? At what temperature should the air be measured in making the volume determination? To what compressor speed does the rated capacity apply? And the most vital question of all: Can capacity be uniquely defined so that it is not a function of any test condition? A definition of capacity which will meet the requirements is believed to be as follows:

The capacity shall be X cubic feet of air per hour at Y pounds per square inch discharge pressure, 68 degrees F. discharge temperature, and Z R. P. M. of compressor with suction conditions corrected to 68 degrees F. temperature, 29.92 inches of mercury barometric pressure, and 36 per cent relative humidity.

Quite naturally the compressor output will vary from day to day, depending on existing conditions at the time. The method of calculation should be such as to correct the performance to standard conditions. The corrected performance should give an accurately reproducible result for compressor output in accordance with the complete definition of capacity.

Some methods of computation in use, wherein correction factors are omitted or incorrectly applied, yield capacity results which are not independent of the test conditions.

There are certain correction factors to be applied in the calculation of air compressor performance. These factors include corrections for:

(*a*) Discrepancy between actual and specified conditions of intake pressures and temperatures.

(*b*) Discrepancy between actual and specified humidity conditions.

(*c*) Discrepancy between actual and specified compressor speeds.

(*d*) Discrepancy between actual and specified discharge conditions of pressures and temperatures.

These correction factors will be briefly discussed.

DEVIATION FROM PERFECT GAS LAWS.

It is generally well appreciated that the behavior of actual gases under varying pressures and temperatures is different from that predicted by the perfect gas laws. The density of air at 3000 pounds per square inch pressure and 70 degrees F. temperature, when calculated from perfect gas laws, will be approximately 3.4 per cent in error when compared to results obtained in computation from experimental data. Likewise, the value of PV for air at 3000 pounds per square inch and 140 degrees F. temperature as calculated by the gas laws, will be approximately 5.5 per cent in error compared to the experimental value of PV. The magnitude of these errors indicates the desirability of recognizing air as an imperfect gas in capacity calculations.

One of the criteria of a perfect gas is its conformity of behavior to the familiar formula

$$PV = NRT$$

where P, V and T are absolute pressure, volume and absolute temperature; N is the number of gram molecules of the gas under consideration; and R is the universal gas constant. According to the kinetic theory a gas is made up of rapidly moving colliding molecules and will conform to the perfect gas equation only if:

(1) There is no mutual attraction between molecules of gas.

(2) The size of molecules is infinitely small as compared to their mean free path.

A real gas will not comply with these conditions, for in a real gas the molecules are of appreciable size and internal work is required to separate them after collision. There are a number of thermodynamic equations of state which attempt to account for the deviation in behavior of a real gas from the ideal gas. The equations of van der Waals, Clausius, Dieterici and Berthelot are perhaps the best known and most nearly approximate the behavior of gases determined experimentally. The equation of van der Waals is typical and is of the form:

$$\left(P + \frac{a}{V^2}\right)(V - b) = NRT$$

Here, P V N R and T have the meanings given above; a and b are constants. The term $\frac{a}{V^2}$ is intended to compensate for the diminishing of observed pressure caused by the cohesive force between molecules. The term b is intended to compensate for the actual volume occupied by gas molecules. This equation has been found to conform to experimental results only in a qualitative way and hence does not accurately predict imperfect gas behavior. This is probably due to an insufficient number of constants to express the complex relationship. Also there appears indication that a and b are actually some function of pressure and volume and hence not constants. Other equations likewise fail to give close agreement with experiment. The deviation of air from perfect gas laws has been experimentally determined by Amagat.*

A series of curves derived from Amagat's data are given in Figure 1. These curves express the values of $K = \frac{459.6}{T} PV$ as a function of P for various conditions of constant temperature. Values of K for intermediate temperatures may be obtained by straight line interpolation. The correction factor K, as determined from these curves, may be used in the equation $PV = KWRT$ for proper values of P and T. In this equation W should be expressed in pounds, P in pound per square foot, V in cubic feet and T in degrees Fahrenheit absolute. The value of R for air when these

* Mémoires sur l'elasticité et la dilatabilité des fluides jusqu'aux très hautes pressions. Annales de chemie et de physique, page 68, vol. 29, 1893.

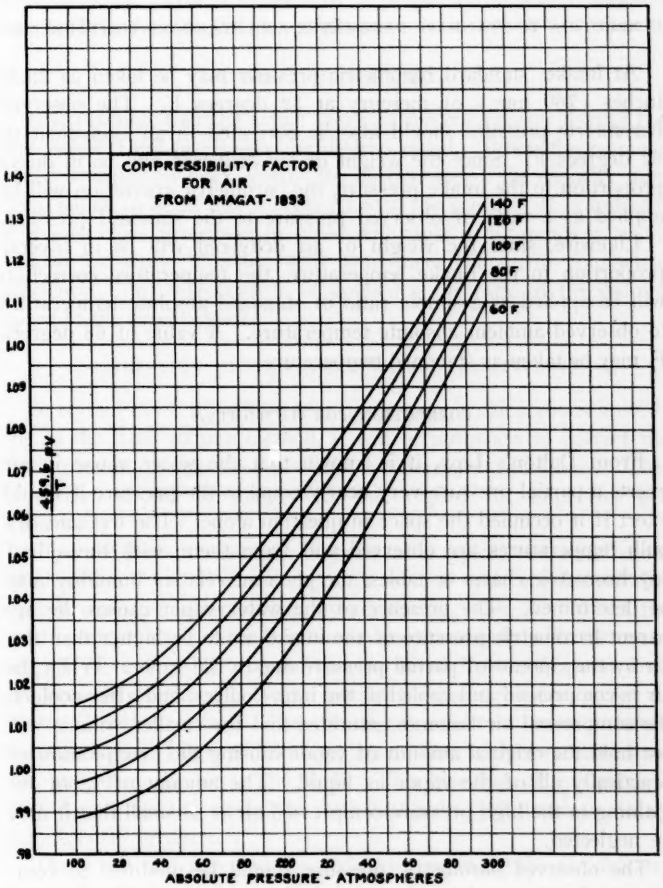

COMPRESSIBILITY FACTOR
FOR AIR
FROM AMAGAT-1893
140 F
120 F
100 F
80 F
60 F
1.14
1.13
1.12
1.11
1.10
1.09
1.08
1.07
1.06
1.05
1.04
1.03
1.02
1.01
1.00
.99
.98
100
20
40
60
80
200
20
40
60
80
300
ABSOLUTE PRESSURE - ATMOSPHERES

FIGURE 1.

units are used may be taken as 53.34. The application of this correction factor may be seen in the sample calculations to follow.

CORRECTION TO STANDARD BAROMETER AND STANDARD TEMPERATURE.

At intake, standard barometric pressure may be taken as 29.92 inches (760 mm.) of mercury at 32 degrees F. The observed barometric pressure should also be corrected to an equivalent of 32 degrees F. Since the weight of air delivered will be in direct proportion to the intake pressure, the barometric correction will be applied as a ratio of observed pressure to the standard pressure.

Likewise, since the weight of air delivered will be in inverse proportion to the intake temperature, the temperature correction will be applied as a simple ratio of standard absolute temperature to observed ambient absolute temperature. A value of 68 degrees F. may be taken as standard temperature.

CORRECTION FOR HUMIDITY.

From Dalton's Law, it is known that the water vapor in air exerts a partial pressure very nearly equal to the pressure it would exert if it occupied the space in question alone. The wet and dry bulb temperatures are observed and from them, with the aid of psychrometric charts or tables, the per cent relative humidity may be determined. The presence of the water vapor causes the apparent barometric pressure of the intake air to be higher than dry air by the amount of partial pressure due to the vapor. When the air is compressed and cooled in the inter-coolers, and after-coolers, the compressed air becomes saturated and the smaller volume will not hold the original amount of vapor causing the precipitation of practically all of the vapor as liquid. The amount of vapor remaining in the high pressure compressed air is so small that it may be neglected.

The observed barometric pressure should be modified to compensate for the partial pressure of the water vapor. The question is, should computations be based on dry air or on some arbitrary humidity percentage? It is entirely unlikely that a compressor will ever handle completely dry air. It would be more nearly representative of average performance conditions if an average value of

humidity were selected as standard. Moss,* in his formulae for flow of air from nozzles has assumed "normal air" to be 36 per cent humidity. This is believed to be proper and has been adopted as a basis in the calculations to follow.

CORRECTION FOR SPEED.

It is probable that the actual speed of the compressor during the test run will not be exactly the desired speed. When the running speed is within five per cent of the desired speed, the capacity may be assumed to be in direct proportion to the speeds. It is unlikely that such close agreement in speed will cause a difference in valve leakage or piston blow-by, with consequent change in volumetric efficiency. The correction to be applied is the ratio of the desired speed to the actual speed.

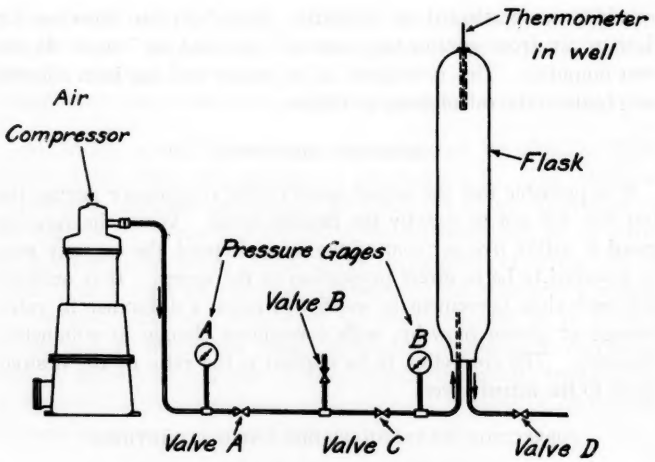
CORRECTION TO STANDARD DISCHARGE CONDITIONS.

In certain cases, as will be seen in the sample calculations according to the flask volume method, it will be necessary to correct the flask content to specified pressure temperature conditions. This correction should be made with due regard for imperfect gas behavior.

Figure 2 gives a schematic diagram of a typical test installation. With valves C and D closed and with valve B open, the desired discharge pressure is established by regulating valve A. When the trial run is to start, valve B is closed simultaneously with the opening of valve C. The elapsed time is noted from this point until the pressure on gauge B is approximately equal to that on gauge A. Valve C is then closed. The volume or weight of water removed through valve D is noted and applied to obtain the net volume or net weight of the air in the flask. This bleeding off of water is best done after the air in the flask has been bled down to nearly atmospheric pressure.

There are two accepted methods of computing the air delivery using the discharge flask. One is based on the net weight of the air in the flask and the other on the net volume. In determining the net weight the flask is weighed before and after charging, with allowance made for the weight of water in the flask. In determin-

* Measurement of Flow of Air and Gas with Nozzles. A. S. M. E. Transactions, Vol. 50, 1928, A. P. M.-3.



Schematic Diagram
Flask Method of Testing
High Pressure Air Compressor.

FIGURE 2.

ing the net volume, the volume of the flask must be known and allowance made for the volume of water removed. This latter method requires a determination of simultaneous values of temperature and pressure in the flask after charging. This may involve inaccuracy due to the difficulty of determining the average temperature of the flask content. If there is definite assurance that no leakage exists, the readings of temperature and pressure may be taken after equilibrium have been established.

Another method of measuring discharge is by means of a receiver chamber in which a calibrated nozzle allows the air to discharge to the atmosphere. This method involves the discharge coefficient of the nozzle which may be uncertain or difficult to determine accurately. This method, however, may be conveniently used where high accuracy is not desired or where low pressures are being dealt with.

When the air intake to the compressor is located so that the air to the suction may be preheated, some confusion may arise as to the proper location for measuring the intake temperature. Two points of view are expressed; one, that intake temperature should refer to the temperature indicated on a thermometer placed in the immediate vicinity of the compressor first stage suction opening, and another, that the intake temperature should be considered as the temperature of the air in the vicinity of the compressor and available for the intake. Consider that air at a certain temperature and pressure, and hence at a certain density, is available in the vicinity of the compressor. If in introducing this air into the first stage cylinder the density is decreased, this decrease is directly effective in lowering from an optimum the weight of compressed air delivered. In an actual compressor this decrease in density may take place in two ways; the air may be preheated by being compelled to pass over hot surfaces in entering the first stage cylinder, or the air may experience a pressure drop in passing through the intake screen, muffler and suction passages. It is obvious that in a sound design any decrease of the air density in the first stage cylinder compared to available air, because of air preheating or suction restriction, should be charged as a loss and recognized as tending to lower the volumetric efficiency. If in capacity calculations the temperature of the air actually in the intake pipe is used, the effect of preheating in lowering the efficiency is not taken into account and the calculations lie on a false basis. For these reasons the intake temperature is considered as equivalent to ambient temperature in capacity calculations.

If the differential between the ambient temperature and actual suction temperature is not constant with changing ambient temperature or circulating water temperature, which will affect the jacket temperature, the reproducibility of results will he affected. However, the differential generally will be small and the change in differential with changing conditions may be disregarded.

CONCLUSIONS.

1. The capacity of an air compressor, at a specified speed, should be expressed in cubic feet per hour at specified standard temperature and pressure conditions of intake and discharge air.

2. Calculations should take into account the deviation of the behavior of air from the perfect gas laws.

3. The weight method of determining compressor output is to be preferred.

SAMPLE CALCULATIONS.

The data used in the following calculations approximate data obtained on actual test but should be considered as illustrative only and not as applying to the performance of any particular compressor.

Method of Calculation Using Net Weight of Flask.

Observed Data.

Speed—454.1 R. P. M.

Barometer—30.16 inches Hg. at 32 degrees F.

Ambient Air { Dry Bulb—82.5 degrees F. / Wet Bulb—74.5 degrees F.

Net weight of air delivered—130.3 pounds.

Elapsed time of trial—28.33 minutes.

Discharge pressure at compressor—3000 pounds per square inch.

Derived Data.

Relative humidity—69 per cent (Marks Handbook, 3rd Ed., page 352).

Saturated vapor pressure at 82.5 degrees F.—1.116 inches Hg. (Marks & Davis Steam Table).

K = 1.034 (Figure 1, Corresponding to P = 205 atmospheres and T = 68 degrees F.

Rated speed—450 R. P. M.

Calculation for Air Density at Standard Temperature at Discharge.

$$W = \frac{PV}{KRT} = \frac{3014.7 \times 144 \times 1}{1.034 \times 53.34 \times 527.6}$$

W = 14.92 pounds per cubic foot at 3000 pounds per square inch gauge and 68 degrees F.

Discharge Volume at 3000 pounds per square inch and 68 degrees F.

$$Q = \frac{\text{Net weight discharged}}{\text{Standard density} \times \text{Elapsed time of trial}}$$

$Q = \frac{130.3 \times 60}{14.92 \times 28.33} = 18.50$ cubic feet per hour at 3000 pounds per square inch and 68 degrees F.

Correction for Speed.

$$Q_1 = \frac{18.50 \times \text{rated speed}}{\text{Observed speed}}$$

$Q_1 = \frac{18.50 \times 450}{454.1} = 18.33$ cubic feet per hour at 3000 pounds per square inch, 68 degrees F. and 450 R.P.M.

Correction for Ambient Temperature at Intake.

$$Q_2 = \frac{18.33 \times \text{absolute ambient temperature}}{\text{Absolute standard temperature}}$$

$Q_2 = \frac{18.33 \times (459.6 + 82.5)}{(459.6 + 68)} = 18.83$ cubic feet per hour at 3000 pounds per square inch, 68 degrees F., 450 R.P.M. and 68 degrees F., intake temperature.

Correction to Standard Humidity of 36 per cent and Standard Barometric Pressure of 29.92 inches Hg. at intake

(Observed humidity — Standard humidity) × Saturated Vapor Pressure = excess partial pressure.

(0.69 — 0.36) × 1.116 = 0.37 inches Hg. excess partial pressure.

Barometer corrected for vapor pressure

30.16 — 0.37 = 29.79 inches Hg.

$$Q_3 = \frac{18.83 \times \text{Standard barometric pressure}}{\text{Corrected observed barometric pressure}}$$

$Q_3 = \frac{18.83 \times 29.92}{29.79} = 18.91$ cubic feet per hour true com-

pressor capacity at 3000 pounds per square inch and 450 R.P.M. corrected to standard conditions.

Method of Calculation Using Net Volume of Flask.

Observed Data.

Compressor speed—429 R. P. M.

Barometer—30.21 inches Hg. at 32 degrees F.

Ambient Air { Dry Bulb—80 degrees F. / Wet Bulb—71 degrees F.

Flask conditions at end of run—P = 2950 pounds per square inch. T = 130 degrees F.

Elapsed time of run——29.15 minutes.

Net volume of flask—9.99 cubic feet.

Derived Data.

Relative humidity—65 per cent (Marks Handbook, 3rd Ed., page 352).

Saturated pressure at 80 degrees F.—1.029 inches Hg. (Marks and Davis Steam Tables).

$K_1 = 1.056$ (Figure 1, Corresponding to 201 atmosphere and 130 degrees F.).

$K_2 = 1.034$ (Figure 1, Corresponding to 205 atmosphere and 68 degrees F.).

Rated speed—450 R. P. M.

Capacity Without Correction.

$$Q_1 = \frac{\text{Net volume}}{\text{Elapsed time}}$$

$$Q_1 = \frac{9.99 \times 60}{29.15} = 20.56 \text{ cubic feet per hour at 2950 pounds per square inch gauge and 130 degrees F.}$$

Correction to Pressure of 3000 pounds per square inch and 68 degrees F. at Discharge.

$$\frac{P_1 V_1}{K_1 RT_1} = \frac{P_2 V_2}{K_2 RT_2} \text{ (Gas laws)}$$

$$V_2 = \frac{P_1 V_1 K_2 RT_2}{P_2 K_1 RT_1}$$

$$V_2 = \frac{2950 \times 20.56 \times 1.034 \times 527.6}{3014.7 \times 1.056 \times 589.6}$$

(Note that gauge rather than absolute pressure is used in the numerator of this expression since the flask was filled with air under one atmosphere pressure at the start of the trial.)

Hence $Q_2 = V_2 = 17.63$ cubic feet per hour at 3000 pounds per square inch gauge and 68 degrees F.

Correction to Ambient Temperature of 68 degrees F. at Inlet.

$$Q_3 = \frac{17.63 \times \text{Observed absolute temperature}}{\text{Standard absolute temperature}}$$

$Q_3 = \frac{17.63 \times 539.6}{527.6} = 18.03$ cubic feet per hour at 3000 pounds per square inch, 68 degrees F. and 68 degrees F. intake temperature.

Correction for Speed.

$$Q_4 = \frac{18.03 \times \text{rated speed}}{\text{Observed speed}}$$

$Q_4 = \frac{18.03 \times 450}{429} = 18.91$ cubic feet per hour at 3000 pounds per square inch, 68 degrees F., 450 R.P.M. and 68 degrees F., intake.

Correction to Standard Humidity of 36 per cent and Standard Barometer of 29.92 inches Hg.

$(0.65 - 0.36) \times 1.029 = 0.30$ excess partial pressure.

Barometer corrected for vapor pressure

$30.21 - 0.30 = 29.91$ inches Hg.

$$Q_5 = \frac{18.91 \times \text{Standard barometric pressure}}{\text{Observed barometric pressure}}$$

$Q_5 = 18.91 \times \frac{29.92}{29.91} = 18.91$ cubic feet per hour true compressor capacity at 3000 pounds per square inch and 450 R.P.M. corrected to standard conditions.

BALANCING PROPULSION SHAFTING.

By RICHARD H. TINGEY.*

This article presents a method of dynamically balancing hollow line and propeller shafts such as those used on naval vessels. The significant feature of the method is that, by taking steps to correct unbalance during the process of machining the shaft, it is dynamically balanced when finished without need for attached weights, eccentric sleeves or such corrective devices.

BASIS OF METHOD.

The principle utilized can be explained by the consideration of a disc of homogeneous material with a circular hole eccentric a small amount w from the outer circumference as shown by full lines in Figure 1. Since the center of gravity G of this body does not coincide with the center O of the outer circle, a centrifugal force will be exerted by the body when it is rotated about O. At a given angular speed, centrifugal force is proportional to the product of the rotating weight and the distance from the center of rotation to the center of gravity of the weight. Find this product for the body shown in Figure 1. Let

W_1 = Weight of the body
W = Weight of the material removed from the hole
$W_1 + W$ = Weight of the solid disc bounded by the outer circumference.

Equating the moment $W_1 v$ of the whole body about O to the sum of the moments of its parts about O:

$$W_1 v = (W_1 + W).\ O - Ww$$
$$W_1 v = -Ww \qquad (1)$$

This states that the centrifugal force is proportional to Ww and is in the opposite direction from the line O O_H representing the

* Assistant Engineer, Fore River Plant, Bethlehem Shipbuilding Corp., Quincy, Mass.

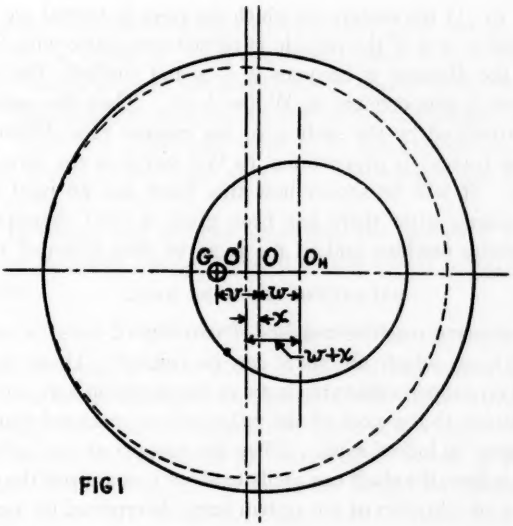


FIG 1

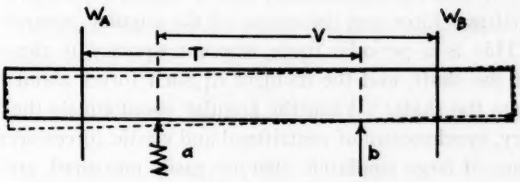


FIG. 2

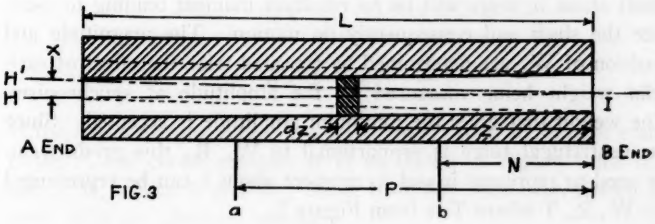


FIG. 3

distance w. If the centers on which the piece is turned are shifted an amount x, and if the outside is turned concentric with the new centers, the distance w becomes $w + x$ and similarly the centrifugal force is proportional to $W\ (w + x)$. Then the centrifugal force introduced by the shift x of the centers (the difference of the above forces) is proportional to Wx and is in the direction of the shift. It will be shown how this force can be used to correct unbalance after there has been given a brief description of the balancing machine and of the required data obtained from it.

BALANCING MACHINE DATA.

The balancing machine consists of two sets of rolls (a and b of Figure 2), on which the shaft can be rotated. These rolls are mounted on supports that are elastic in one direction only, in a plane perpendicular to the axes of the rolls, and so arranged that either support may be locked solid. When the support at b is locked and that at a is free, the shaft can oscillate about b as a pivot, the natural frequency of vibration of the system being determined by the elastic characteristics of the support and by the mass of the shaft. When the unbalanced shaft is rotating, the component of the centrifugal force in the direction of elasticity of the support is the product of the centrifugal force and the cosine of the angular position of the shaft. This is a periodic force whose frequency is the angular speed of the shaft, and the moment of such forces about b tends to oscillate the shaft. When the angular speed equals the natural frequency, synchronism of centrifugal and elastic forces occurs and oscillations of large amplitude, that are easily measured, are set up.

If the moment about b of the centrifugal force due to a weight W_A attached at a radius R_A in a convenient plane A is equal and opposite to the moment of the unbalanced forces of the shaft about b, there will be no resultant moment tending to oscillate the shaft and consequently no motion. The magnitude and position of W_A are determined by repeated trial, the effect of each trial weight being measured by the amplitude at synchronism. The weight that gives the minimum amplitude is accepted. Since the centrifugal force is proportional to $W_A\ R_A$ this product can be used to represent it and its moment about b can be represented by $W_A R_A\ T$ where T is from Figure 2.

Then with support a clamped and b free there can be found a weight W_B at radius R_B in a convenient plane B (not coinciding with A) which reduces the amplitude at synchronism to a minimum. This indicates that the moment of the centrifugal force due to W_B about a is equal and opposite to the moment of the unbalanced forces of the shaft about a. The moment can be represented by $W_B R_B V$.

The following two sections show how the balancing effects of these two moments can be replaced by the effects of turning center shifts.

EFFECT OF CENTER SHIFTS.

Figure 3 shows a shaft whose A end turning center is shifted from H to H′ an amount x and whose B end center remains at I. The outside is turned to a cylinder on these new centers. The shift of the center of rotation at an intermediate point, a distance z from the B end, is xz/L. If W is the weight of material taken from the whole bore, the weight taken from an elementary cylinder of length dz is Wdz/L. The centrifugal force introduced by the shift of the center of this element is proportional to (Wdz/L) (xz/L) and the moment of this force about a is (Wdz/L) (xz/L) (z-P). Integrate this over the whole length of the shaft to find M_{ax} the moment about a due to the shift x.

$$M_{ax} = \frac{Wx}{L^2}\int_0^L (z^2 - z\,P)\,dz$$

$$= Wx\left(\frac{L}{3} - \frac{P}{2}\right) \quad . \quad . \quad . \quad (2)$$

M_{bx} the moment about b due to the shift x, is found similarly

$$M_{bx} = \frac{Wx}{L^2}\int_0^L (z^2 - z\,N)\,dz$$

$$= Wx\left(\frac{L}{3} - \frac{N}{2}\right) \quad . \quad . \quad . \quad (3)$$

If the B end turning center is shifted from I to I′ an amount y and if the A end center remains at H as shown in Figure 4 the moments M_{ay} about a and M_{by} about b of the forces introduced by their shift are obtained similarly.

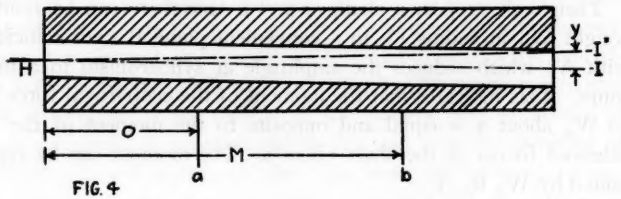


FIG. 4

$$M_{ay} = Wy\left(\frac{O}{2} - \frac{L}{3}\right) \quad . \quad . \quad . \quad (4)$$

$$M_{by} = Wy\left(\frac{M}{2} - \frac{L}{3}\right) \quad . \quad . \quad . \quad (5)$$

CENTER SHIFTS TO REPLACE TRIAL WEIGHTS.

By a suitable shift of turning centers x_1 and y_1 it is possible to introduce a moment about b equal to that of $W_A R_A$ and, at the same time, to introduce no moment about a. To make the moments about b equal, using (3) and (5).

$$W_A R_A T = Wx_1\left(\frac{L}{3} - \frac{N}{2}\right) + Wy_1\left(\frac{M}{2} - \frac{L}{3}\right)$$

If no moment about a is to be introduced, using (2) and (4).

$$O = Wx_1\left(\frac{L}{3} - \frac{P}{2}\right) + Wy_1\left(\frac{O}{2} - \frac{L}{3}\right)$$

These simultaneous equations are solved for x_1 and y_1 giving:

$$x_1 = \frac{W_A R_A T}{W K}\left(\frac{L}{3} - \frac{O}{2}\right) \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$y_1 = -\frac{W_A R_A T}{W K}\left(\frac{P}{2} - \frac{L}{3}\right) \quad . \quad . \quad . \quad . \quad (7)$$

$$K = \left(\frac{L}{3} - \frac{N}{2}\right)\left(\frac{L}{3} - \frac{O}{2}\right) - \left(\frac{P}{2} - \frac{L}{3}\right)\left(\frac{M}{2} - \frac{L}{3}\right) \quad (8)$$

When the balancing machine is operated with the b rolls clamped these turning center shifts have the same effect as the trial weight W_A and when the a rolls are clamped they have no effect.

Similarly the shifts x_2 and y_2 are equivalent to W_B with the a rolls clamped and have no effect with the b rolls clamped.

$$x_2 = -\frac{W_B R_B V}{WK}\left(\frac{M}{2} - \frac{L}{3}\right) \quad . \quad . \quad . \quad . \quad (9)$$

$$y_2 = \frac{W_B R_B V}{WK}\left(\frac{L}{3} - \frac{N}{2}\right) \quad . \quad . \quad . \quad . \quad (10)$$

If the shifts x_1, y_1, x_2 and y_2 are made simultaneously they have the same effect as W_A when operating with the b rolls clamped and the same effect as W_B when operating with the a rolls clamped. As stated in the discussion of the balancing machine, the application of W_A reduces the resultant moment of all centrifugal forces about b to zero and W_B reduces those about a, to zero. Therefore the shifts x_1, y_1, x_2 and y_2 make these moments zero. By a principle of elementary mechanics, if the moments of a system of forces about each of two points are zero, the resultant force is zero and the moment of the forces about any point is zero. Since these are the conditions required for dynamic balance, now the shaft is satisfactorily balanced.

It will be noticed that x_1 and y_1 are in the plane of the trial weight W_A and x_2 and y_2 are in the plane of W_B. When these shifts are made simultaneously, x_1 and x_2 can be combined vectorially into a single shift x on the A end and y_1 and y_2 into a shift y on the B end. In equations (6), (7), (9) and (10) the positive sign indicates that the center is to be shifted towards the trial weight from which the shift is derived and the negative sign indicates a shift away from the weight.

It simplifies the calculation slightly if the set-up of the balancing machine, shaft and discs is made symmetrical. Then since $M = P$, $O = N$ and $T = V$ the coefficient of $W_A R_A$ to find x_1 is equal to that of $W_B R_B$ to find y_2 and the coefficient of $W_A R_A$ to find y_1 is equal to that of $W_B R_B$ to find x_2. A further simplification results if the rolls are placed one-third of the length of the shaft from its ends. In this case since $M = P = 2L/3$ the shifts x_2 and y_1 become zero making $x = x_1$ and $y = y_2$ thus eliminating the graphical combination of components required by the general case. However, if there are shafts of various lengths in the shop at the same time, as is usually the case, it is not convenient to apply this simplification to all of the sizes because it involves changing

the balancing machine set-up many times, wasting more time than is saved in the calculation.

PROCEDURE.

The procedure described below was developed to produce a satisfactory shaft system with the minimum deviation from the usual machining process. First a light cut is taken off the shaft forging, rough turned and bored as it comes to the machine shop, to insure that its outer surface is cylindrical and concentric with the turning centers throughout its length. Two short lengths, usually located equidistant from the ends, are finished smooth to accommodate the balancing machine rolls. Then the shaft, with two discs symmetrically located between the rolls and the ends for the attachment of trial weights, is placed on the machine and the two trial weights W_A and W_B are found. The magnitude and position of the resultant shifts x and y are calculated and given to the balancing machine operator who scribes a line on each end of the shaft at the designated angular positions. The shaft is returned to the lathe where the balancing machine operator directs the shifting of the centers which is accomplished by scraping. A series of cuts is made, removing a part of the excess material, the last cut being light to insure that the shaft is straight. The lengths to accommodate the rolls are finished smooth and the above process repeated. It is considered undesirable to finish the shaft after one correction because the process of shifting the centers is subject to some inaccuracy and because any inaccuracy or error is irreparable if the shaft is down to size except by the use of corrective devices. Usually a shaft is corrected three times in this manner and, after finishing, is placed on the machine to demonstrate that it is balanced. A shaft is considered satisfactorily balanced if the sum of the centrifugal forces that would be produced at full power revolutions by the weights (applied at the ends) necessary to balance it, is not greater than two per cent of the weight of the shaft.

SAMPLE CALCULATION.

The following sample calculation illustrates the method. Figure 5 gives the data supplied by the balancing machine operator and

Figure 6 shows the derived lengths required for the calculation. From the plan dimensions the weight W of the metal removed from the bore is found to be 137,500 ounces. Substituting in (8), (6), (9), (7) and (10):

$$K = \left(\frac{476.5}{3} - \frac{104.2}{2}\right)^2 - \left(\frac{372.3}{2} - \frac{476.5}{3}\right)^2 = 10{,}635$$

$$x_1 = +\,43.5\,\frac{14.5 \times 308.8}{137{,}500 \times 10{,}635}\left(\frac{476.5}{3} - \frac{104.2}{2}\right) = +\,.0142$$

$$x_2 = -\,61.5\,\frac{14.5 \times 308.8}{137{,}500 \times 10{,}635}\left(\frac{372.3}{2} - \frac{476.5}{3}\right) = -\,.0051$$

$$y_1 = -\,43.5\,\frac{14.5 \times 308.8}{137{,}500 \times 10{,}635}\left(\frac{372.3}{2} - \frac{476.5}{3}\right) = -\,.0036$$

$$y_2 = +\,61.5\,\frac{14.5 \times 308.8}{137{,}500 \times 10{,}635}\left(\frac{476.5}{3} - \frac{104.2}{2}\right) = +\,.0200$$

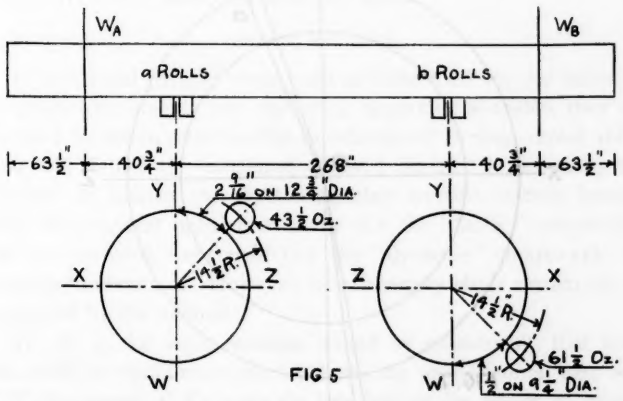


FIG 5

The angular positions of the weights W_A and W_B are laid out on Figure 7 according to the data from Figure 6. Then x_1 is laid off to scale from O in the direction of W_A becoming the line Oa, and x_2 is laid off to scale from a in the opposite direction from W_B becoming the line a b. Then the line Ob represents to scale the

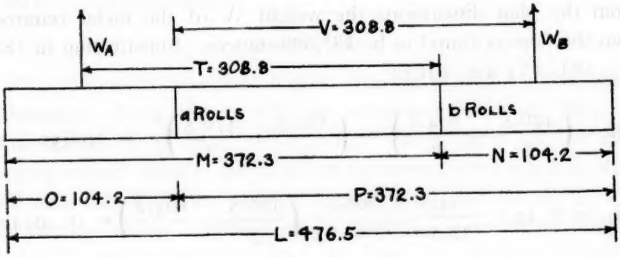

W_A
V=308.8
W_B
T=308.8
a ROLLS
b ROLLS
M=372.3
N=104.2
O=104.2
P=372.3
L=476.5

FIG. 6

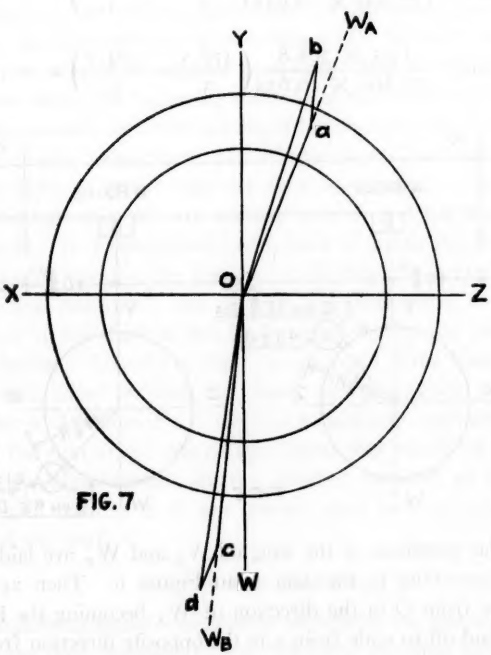

Y
b
W_A
a
O
X
Z
c
W
d
W_B

FIG. 7

resultant center shift x on the A end. Similarly y_2 is represented by Oc, y_1 by cd and y, the resultant shift of the center on the B end, by Od.

After these shifts are made and the shaft is turned down to finished dimensions, it is placed on the balancing machine and shows some unbalance. With the b rolls clamped, a piece of putty on the A end of the shaft at a convenient radius serves as a trial weight that can be varied to reduce the amplitude approximately to a minimum. Similarly putty can be so used on the B end when the a rolls are clamped. The sum of these two balancing weights is 3 pounds at a radius of 7⅛ inches. The allowable weight w is found by equating its centrifugal force to two per cent of the shaft weight. The full power revolution per minute of this shaft are 412 and the shaft weight is 14,000 pounds.

$$4\pi^2 \left(\frac{412}{60}\right)^2 \frac{7.125}{12} \frac{w}{32.2} = 14{,}000 \times .02$$

$$w = 8.1 \text{ pounds}$$

The shaft is well within the allowable limit.

STATIC BALANCE.

If horizontal parallel ways, such as those used for the balancing of propellers, are the only balancing apparatus available, they can be used to obtain data for the calculation of turning center shifts that will reduce the unbalance of most shafts. It is usually impossible to balance the shaft completely by this method because only the resultant unbalanced force (*i.e.* the "static" component) can be corrected, the couple (*i.e.* the "dynamic" component) remaining undetected. However, a satisfactory shaft system can be produced by the method.

$W_o R_o$ is the static moment, found by experiment, that holds the shaft in equilibrium when it is resting on horizontal bars with G O horizontal, G O being the line joining the center of gravity and the center of the outer circumference as shown in Figure 8. To find the turning center shifts x that make the center of gravity of the shaft lie on the central axis of its outer surface after machining, equate $W_o R_o$ and Wx (where W is the weight of the metal removed from the bore). Then

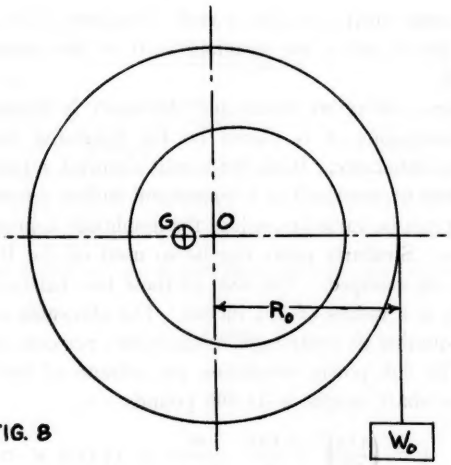


FIG. 8

$$x = \frac{W_o}{W} R_o$$

To correct the unbalance the centers on the ends are moved a distance x towards the balancing weight W_o and the shaft is machined on these centers.

A GRAPHICAL MEANS OF CHECKING FUEL OIL HEATER PERFORMANCE.

By Lieut. Comdr. R. L. Hicks, U. S. N., Member, and H. E. Bethun.

Fuel oil heaters on board Naval ships are installed in a capacity sufficient for maximum power conditions. The fact that these ships habitually operate at a fraction of maximum power, means that the oil heaters are seldom operated at their maximum designed rate. Under these conditions, where the demand on the heater is low, there is a probability that the heater may be giving a poor performance, and the fact will not be realized until the demand for maximum capacity occurs—when the heater may be found inadequate for the duty demanded.

To enable the operating engineer to check the operation of his fuel oil heaters under any condition whatever, this chart has been prepared. The chart is applicable to all heaters of the straight, slightly bowed, or V-tube type, without retarders, and with either ⅝ or ¾-inch tubes, and may be used for heaters of a wide range in size and in oil capacity. The chart is not strictly accurate in all respects, in that some of the minor variables have been neglected. For example, in determining the heat transfer rate for a certain oil, at a certain tube velocity, the viscosity of the oil is made the prime variable, and a mean value is taken for minor variables—for example, the specific heat of the oil. The heat transfer rates given are founded on the usual resistance method of determination, and as plotted, are in fair agreement with a great number of fuel oil heater tests available for check. The curves as given are felt to be sufficiently accurate for an excellent check on the performance of a heater.

The method of using these curves is as follows:

The operating engineer should plot a curve of gallons of oil per hour *vs.* pounds of oil per tube per hour for his particular heater.

The number of tubes used in making this plot should be the average number of tubes in each pass of the heater. From this curve the pounds of oil per tube per hour for any oil rate of his heater may be found.

With the pounds of oil per tube per hour, he may then enter Curve A, and for the oil he is using, the heat transfer rate the heater should be giving, is determined. With this heat transfer rate, Curve B is entered, and with the size of the heater, the pounds of oil per hour, the heating steam temperature, and the oil inlet temperature as arguments, the outlet oil temperature that should be obtained, is determined. If the outlet oil temperature actually existing is materially below that indicated, the heater performance is unsatisfactory. The reason for the unsatisfactory performance may be dirty tubes, improper drainage of the heater, air pockets, or any one of another other reasons.

The above curves offer no results that cannot be obtained at any time by computing the heater performance. They do offer a quick and fairly accurate means of computing the performance of the heater by graphical means, and in such a form that it may be used by any one.

CURVE "A"

POUNDS OIL PER TUBE PER HOUR

CURVE "B"

OUTLINE OF RESISTANCE WELDING AND ITS USES.

By BELA RONAY.*

Electric resistance welding equipment is used today in many fabricating shops for a large variety of work. The intelligent use of the medium power multi-armature resistance welders simplifies many design problems and leads to better and less expensive construction.

Resistance welding is a modern version of the blacksmith weld in its essentials; however, it has so many more applications than the crude methods of the smith, that the identification of the underlying principle is somewhat obscured.

"Resistance welding is a method of joining metals under pressure, when heated to the plastic state." This often-used definition of resistance welding would automatically exclude all non-ferrous metals that do not have a transitory plastic state. This unfavorable characteristic of non-ferrous metals does limit the use of resistance welding in certain applications to thin sections only, but such may be successfully welded.

Resistance welding is done with specially built apparatus, consisting of the following parts:

(*a*) *The Transformer*—usually a single phase auto transformer, wound on an annular ring core and equipped with several taps. The primary of the transformer is usually the (220-volt) power line. The secondary taps range from 1.5 to 10 volts, the capacity from 1 to 75 KVA.† Resistance welders built for production work have been built with capacities as high as 3000 KVA for flash welding automobile rear axle housings or body parts. The general utility welder for shop use seldom exceeds 35 KVA capacity.

(*b*) The secondary leads are connected to the working arms or armatures of the welder. The armatures terminate in the

* Welding Assistant, Engineering Experiment Station.

† See Figures 1, 2, 3 and 4.

electrodes. A general utility welding machine may have interchangeable armatures in that the same machine may be set up to do any of the following operations: (1) Spot welding. (2) Butt welding. (3) Seam welding—axial. (4) Seam welding—girthwise.

Spot Welding. The spot welding armature consists of two arms; one is rigidly fixed, while the other, usually the upper one, is pivoted in the manner of a rocker arm. The tips of these arms are the electrodes, which are usually water cooled. These electrodes are made of a heat-resisting copper alloy of good electrical conductivity. The electrodes are shaped to suit not only the gauge of the materials to be joined, but also the contour of work, so that work may be done at joints that require offset electrodes. The pivoted arm is actuated by means of a large ratio lever system, or by pneumatic or hydraulic power, to exert the required pressure on the welds. When the two electrodes are pressed against the outer surfaces of the joint the secondary circuit of the transformer is closed across the electrodes, sending a heavy current through the limited path between them. The current density is so regulated that the metal heats instantaneously to a welding temperature. At this temperature the metal flows readily and the weld is completed when it is felt to be yielding under the electrode. The spots solidify immediately after the opening of the circuit. In the case of welding mild steel the temperature reached with the current on is not in excess of 2600-2700 degrees F.; *i.e.,* it is below the melting point of the metal, as steel welds in its plastic state. No flux is used; however, the metal faces to be welded must be cleaned by pickling or by mechanical means to secure welds free of inclusions. The pressure exerted on the parts to be joined is approximately 5000 p.s.i. for ferrous metals and less for non-ferrous materials.

Non-ferrous metals of thin section which on heating enter into a liquid state without first assuming the transitory plastic state, may be also spot welded by the resistance welding method. When welding such metals the current density is increased to a higher value than would be required for a corresponding thickness of steel and the spot diameter is reduced. This localizes the heat to an even greater extent than is done in the case of steel welding.

The metal between the electrodes actually becomes molten but cannot run out as the surrounding solid metal holds it by capillary attraction as in a small crucible. There is a very slight trace of sloughing on the underside of such welds, testifying to the liquid state of the metal between the electrodes. Certain metals, especially the low temperature alloys, such as aluminum and magnesium, require special chilling fixtures to be attached to the lower or both electrodes, to help heat concentration, especially when the work is done on heavier gauge material requiring a larger spot diameter. The following materials are suitable for spot welding:

All ferrous metals and alloys; monel metal; copper; brass and bronze; tungsten (in wire form); gold, silver and their alloys; German silver, and practically all copper-nickel and nickel-copper alloys.

Strength of spot welds. The strength of spot welds is usually calculated by regarding each individual spot as a rivet in single shear. The spot spacing then depends on the load the joint is to transmit, and an arrangement similar to rivet spacing may be used.

The cross-sectional area of spots varies with the type of equipment used, as well as the condition and temperature of the electrodes used. The best spot welds require the use of clean stock. They are of minimum diameter produced between well pointed electrodes, transmitting a high current density during a minimum current dwell. Such welds are apt to be free of atmospheric contamination and have their full strength throughout their entire cross-sectional area. When the electrodes are blunt, the current density becomes automatically lower, consequently the heat concentration suffers, thus the completion of the weld requires a longer period. If this longer period is not given to the weld, then of course, a partial weld is the result. The strength of a spot welded joint consequently is dependent almost entirely on the condition of the electrodes, the skill and the integrity of the operator. For this reason automatic devices have been added to spot welding machines, that compensate for the human element as far as current regulation is concerned, but still such equipment cannot correct the laxity in keeping the electrodes dressed to the proper shape and diameter. The automatic devices may be best described as energy measuring devices, in so far as they automatically reg-

ulate the energy input during the production of each weld, assuring a minimum dwell period and minimum current delivery for each weld independent of the operator, who is unable to open the circuit or release the pressure until the required current dwell is completed. "Shot welding" is a precision execution of time and current controlled spot welding.

Size of spot welds. On materials up to 30 gauge the spot diameter need not exceed 1/16 inch; up to 20 gauge, 3/32 inch; up to 15 gauge, 1/8 inch; and from 16 to 10, 5/32 inch diameter.

Butt welding. The simplest and most commonly used butt welding armature consists of two vises electrically insulated from one another and mounted on the transformer frame work that is designed similar to a lathe bed.* One of these vises is in a fixed position, while the other is axially slideable like the tail stock of a lathe. The materials to be united by welding are engaged in the two vises so that the ends to be joined are facing one another. These ends are previously cleaned and squared. The vises contain the usually water-cooled electrodes. When the materials to be welded are firmly gripped in the vises the movable one is pushed up against the stationary vise until the ends meet, then the current is short-circuited between the overhanging ends of the pieces to be joined. These ends soon become hot and when they show the right color for welding the operator exerts some further pressure on the lever system that advances the slideable electrodes, thus upsetting the weld.

The electric resistance butt weld is the nearest approach to the original blacksmith weld, and, like that, it is the least efficient. This method, though many times faster than the blacksmith, is not sufficiently fast to permit the completion of the welding without exposing the white-hot joint to atmospheric contamination. The upsetting tends to squeeze out the oxidized edge of the joint and help to ameliorate this condition, but it is not a wholly dependable performance because the upsetting is seldom uniform enough. For this reason butt welds of this type are seldom used for the fabrication of equipment that is seriously stressed in service. Their use is relegated almost entirely to the field of ornamental metal work where the efficiency of the weld or of the joint is of minor

* See Figures 5 and 6.

importance. A modification of the butt weld, the so-called flash weld, is used when joints of reliable strength are to be produced.

The flash welding apparatus is a modified butt welder. In the pressure link system of this method there is a cam incorporated that modifies the straight, forward movement of the sliding vise. First, the sliding vise and with it one member to be joined travels toward the stationary vise and the other member until the ends meet. Then by the action of the cam the ends are slightly separated, whereby an arc is established between them. By further advancing the control lever the motion of the sliding vise is reversed and the ends are again approaching one another at a rate that corresponds to the consumption of the metal by the arc. Finally the movement is accelerated and, as the ends meet and are upset, the current is cut off. During upsetting, the oxidized ends are thoroughly and uniformly squeezed out and the metal within the original section of the weld is pure, though somewhat overheated. These welders, like spot welders, are usually equipped with automatic devices that maintain from weld to weld a constant volume of metal flashed off and permit the repeating of a predetermined current input. The time of flashing, equivalent to the current dwell in spot welding, is also automatically regulated. The output of such flash welders may be reliable and may be considered as a means of fabricating stressed members. The cross-sectional area of the members to be joined is limited by the capacity of the transformers only. Commercial equipment seldom exceeds a capacity of over 300 KVA. Such equipment may handle steel up to 12 square inches in cross-section or non-ferrous metals up to 6-8 square inches in cross-section, depending on the conductivity of the metals to be joined.

Seam welding. Seam welding consists of producing a series of overlapping spot welds. It may be applied to a longitudinal seam or to a girthwise seam. Each type requires a different set of electrodes and driving devices to accommodate the work. The electrodes are wheels instead of the studs used in making individual spot welds.* To actually produce independent overlapping spots in course of the making of a seam weld, it is necessary to interrupt the current with a certain regularity, allowing a suffi-

* See Figures 7 and 8.

cient current dwell to produce a single spot which is somewhat smaller than would be if the same gauge metal were spot welded. The interrupter may be purely mechanical or thermionic tube actuated, the latter being more accurate and apt to give less maintenance trouble than the former.

The seam-welded joints produced by machines equipped with well designed and regulatable interrupters are quite reliable, pressure tight and have a joint efficiency of close to 100 per cent. Such joints are desirable with materials not heavier than 18 gauge. With materials heavier than 18 gauge the strength and tightness of the seam welded joints is no longer reliable for anything but inexpensive mass articles destined for domestic use only. All materials that may be spot welded may be seam welded also.

The advantages of spot welding over mechanical joints:

(1) Saving of weight by reducing the width of overlap and by eliminating bolts, rivets and butt straps.

(2) Neater design—the absence of bolts, nuts, etc., permits a better finish.

(3) Tight, leak-proof joints, that remain leak-proof during the life of the vessel.

(4) Lower cost of construction.

DESIGN CONSIDERATIONS FOR SPOT WELDED CONSTRUCTION.

The average diameter of spots for a given gauge of metal has been already given. In designing a spot welded joint the same considerations hold good as in riveted construction. The free edge of the sheet, *i.e.*, the distance of the center line of spots from the edge of the material, must contain a sufficient volume of metal to resist the load transmitted by a single spot in double shear. (The spot welded area is of course in single shear only.)

A spot welded design always includes a joint, hence a spot welded pressure vessel may develop slight leaks in the same manner as a riveted vessel is apt to do without caulking. However, the thin gauge of the material does not suffice for caulking, therefore spot welded pressure vessels are made leak tight by soldering, a common practice in stainless steel construction. Fuel tanks and even compressed air receptacles have been so constructed by

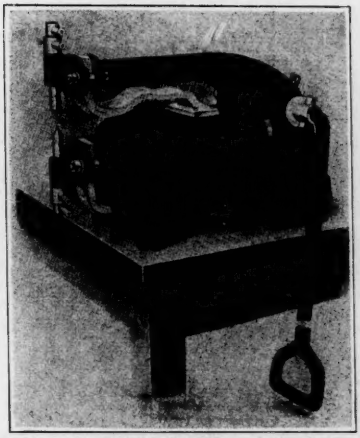
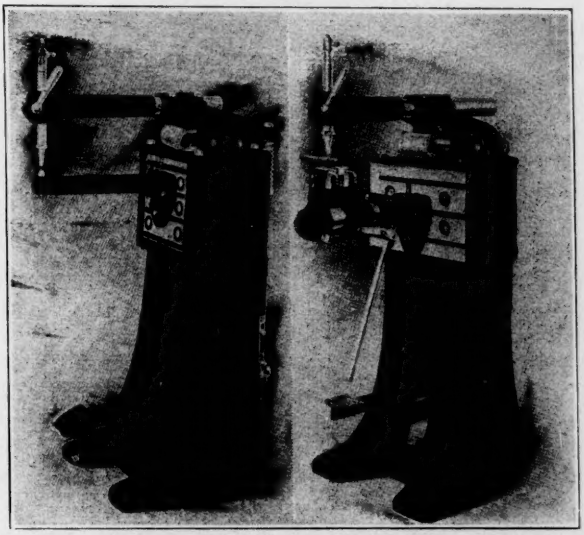


FIGURE 1.—2 KVA CAPACITY SPOT WELDER.



FIGURES 2 AND 3.—25 KVA CAPACITY SPOT WELDERS WITH SPECIAL ARMATURES.

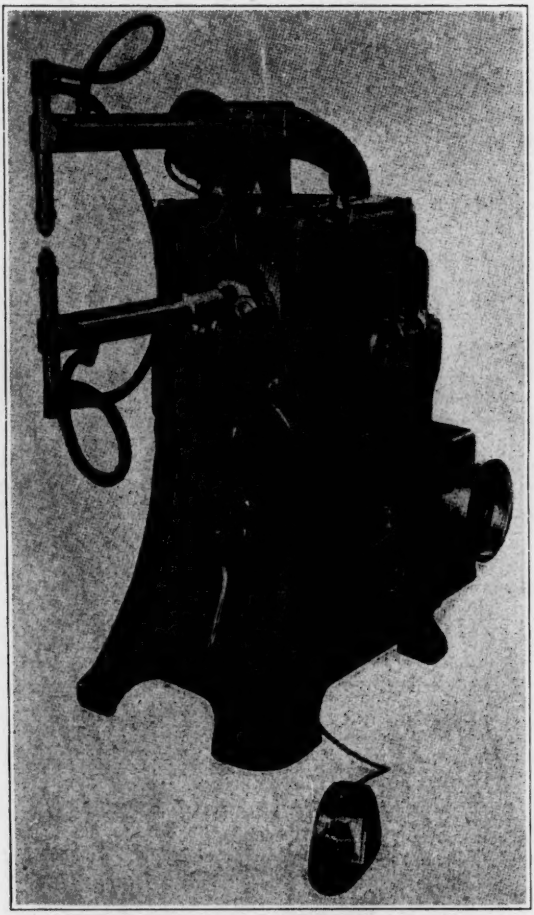


FIGURE 4.—35 KVA CAPACITY SPOT WELDER WITH AUTOMATIC TIME CURRENT RATE REGULATING DEVICE.

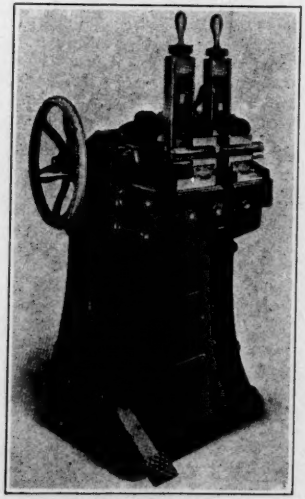


FIGURE 5.—25 KVA CAPACITY BUTT WELDER.

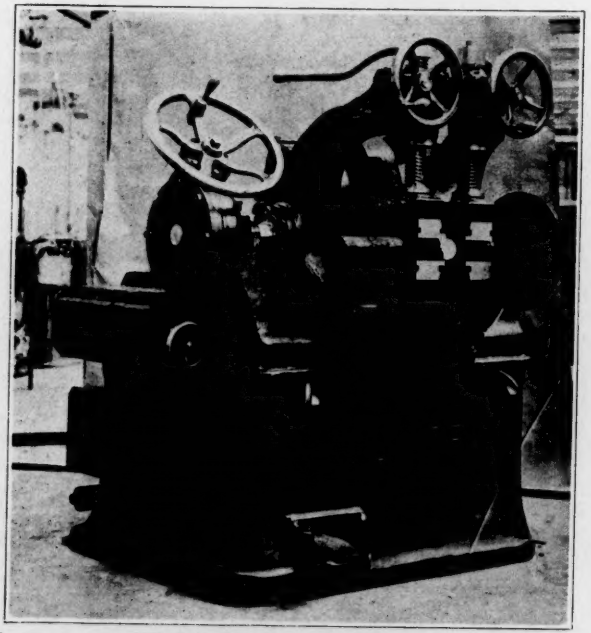


FIGURE 6.—150 KVA CAPACITY SEMI-AUTOMATIC FLASH WELDER.

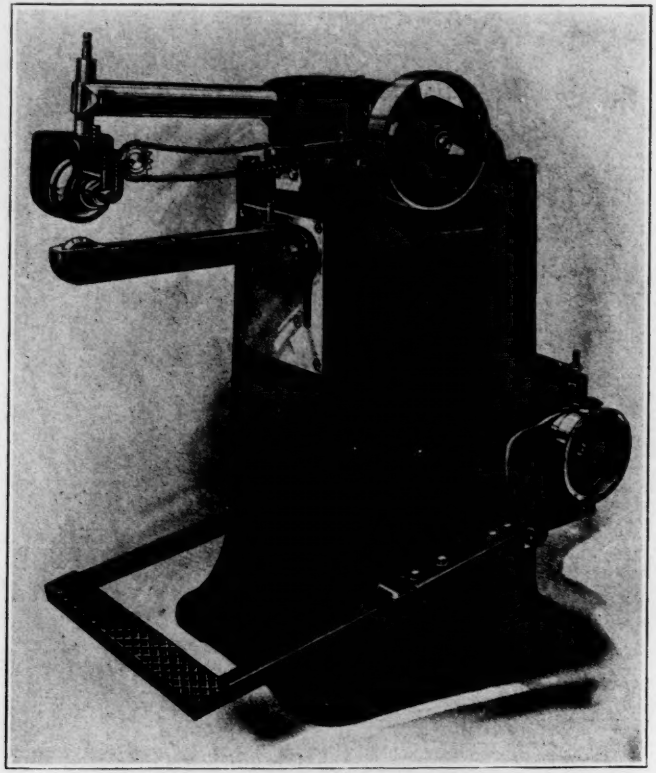


FIGURE 7.—16 KVA CAPACITY SEAM WELDER WITH ARMATURES FOR WELDING LONGITUDINAL SEAMS.

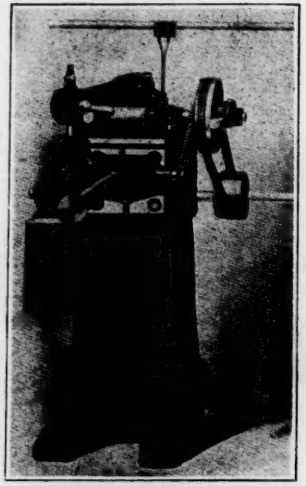


FIGURE 8.—16 KVA CAPACITY SEAM WELDER WITH ARMATURES FOR WELDING CIRCUMFERENTIAL SEAMS.

spot welding. When the gauge of material permits seam welding, its application should be preferred as seam welded joints are practically lapless and they are leak tight. In making a seam welded joint the original overlap is but slightly in excess of twice the thickness of the material. During welding the ridge formed by the overlap is squeezed down to slightly in excess of a single thickness. In many cases a simple grinding operation removes the slight excess metal from the inside of the seam of a welded cylinder, leaving the interiors smooth enough to permit a drive fit of dished heads. This type of construction, resulting in a smooth interior or exterior, or both, is desirable when sanitary requirements prescribe smooth, ridgeless interiors. Seam welding is the proper method of production when a high-class finish is required on the outside, or when the finished cylinder is to be used as a sleeve fitting snugly between an inner core and outer shell, so that both the interior and exterior must be smooth and the cross-section of the cylinder must be a truly concentric annular ring. Seam welding or spot welding offers a very acceptable substitute in the fabrication of articles that would be produced by stamping or deep drawing, if the quantity required would warrant the design and construction of the usually costly dies. Seam welding is the best substitution as it permits practically seamless construction. It should be borne in mind that a seam-welded article may be treated in further work in the same manner as the unwelded metal, *i.e.*, it may be spun or rolled, without any injury to a properly made seam weld. Canisters and other containers reenforced with corrugations, are often made of sheet stock, seam welded, the corrugations being rolled after the completion and surface finishing of the welds.

Spot welded assemblies are not restricted to "shot welded" construction only, in so far that "shot welding" is simply a trade name applied to a controlled process of spot welding. Any manufacturer may equip himself with a similar device and may fabricate reliably welded structures in any metal suitable for spot welding. One of the advantages of spot or seam welded construction is that the welded assembly requires no stress relieving treatment, as with ordinary care it is possible to so set up the work for welding that the order of shrinkage stresses is negligible. Nor is the destruc-

FIGURE 7.—16 KVA CAPACITY SEAM WELDER WITH ARRANGEMENT FOR WELDING LONGITUDINAL SEAMS.

tion of the original grain structure in the weld proper of great consequence, as it is confined to small areas only. Of course all resistance welds are in effect similar to fusion welds in the not-normalized state. The metal within the area of the spot weld is in the "as cast" condition, having a higher tensile strength than the normalized metal, but considerably lower ductility in case of ferrous metals. The pressure exerted on the hot metal does not have the effect of hot working on account of the extremely short duration. Nevertheless, as mentioned before, it is possible to submit seam-welded metal to the same cold working procedure as may be safely carried on with the annealed base metal, because the hardened cross-section is of small dimensions.

COMMENTS ON BUTT AND FLASH WELDS.

In case of butt and especially flash welding, the situation is not the same, by virtue of the fact that this method is applied to an entirely different class of work. The articles that may be produced by flash welding, like valves, tube joints, etc., are subject to not only static but also dynamic loads. Therefore the condition of the metal structure throughout the entire mass is of great importance. Hence, a suitable heat-treatment after welding is often a basic requirement. Whenever the shape or design permits it, a localized heat-treatment may be sufficient, affecting only such areas that were exposed to the heat of welding.

In finishing flash welds, it is customary not to remove the entire reenforcement or upset, that formed at the completion of a butt or flash weld, merely finishing the joint by a trimming of its contours with a forming tool or by grinding. Consequently, the joint efficiency of such welds may appear 100 per cent at most tests, in spite of the fact that the weld efficiency might be considerably below that. In contrasting the joint efficiency to weld efficiency, the following is meant: The cross-sectional area of the metal of 2-inch tubing having a ⅛-inch wall thickness is 0.74 square inch; with the customary reenforcement of 10 per cent at the centerline of the weld, the O.D. is 2.2 inches and the corresponding cross-sectional area is 1.4 square inches. Consequently it is feasible, when a welded joint is tested with the reenforcement left on, for the break to occur in the parent metal as

long as the weld efficiency is but slightly over 50 per cent. When the break occurs outside the joint, the joint efficiency is 100 per cent, though further examination may reveal that the joint has not been fully welded. Hence, testing such joints by static loads only is not reliable. The true condition of the weld may be ascertained by bend or nick break tests and the true evaluation of the joint by fatigue tests. Microscopic examination of the weld zone may reveal the nature of defects in a flash or butt weld, but since such examination is confined to extremely small areas only, it cannot be used for evaluation of the entire joint.

Flash welding has not gained sufficient importance as yet in general engineering practice, as it has been treated as a production method only. However, with the advent of the use of several new alloy steels in the manufacture of fittings for high pressure and high temperature steam systems, it will be soon appreciated as a suitable means of fabricating special shapes in preference to fusion welding. Its use would eliminate the necessity of developing new electrodes and suitable welding technique for each new alloy produced.

A SHORT HISTORY OF THE NAVAL USE OF FUEL OIL.

PART III.

By Lieutenant James E. Hamilton, U. S. N., Member.

THE WAR PERIOD, 1917-1920.

The preceding section brought the fuel oil history up to the war preparation period which is to be included under this section as part of the war period. The previous period logically ends with the dissolution of the Fuel Oil Board.

The fuel oil problem, toward the end of 1916, was developing very nicely. The summary of the preceding section shows that all phases of the subject were under active study. Unfortunately, few of them had reached the stage where war interference would not interrupt progress. This point will be more clearly demonstrated in the events of the post-war period.

The development which was underway as previously described was arrested and superseded by the emergency of war. The problems which arose specifically in connection with fuel oil were ones which were thrust upon the Navy by the particular exigencies of an unusual and certainly not long contemplated type of war for the United States. Being injected into the middle of a war which had developed to the stage where the new ally, the United States, became an immediate reservoir of men, money, and materials, when active planning for the supply of these three essentials had been initiated only a few months before the heavy demand fell, permitted no opportunity for the continuation of any orderly scientific progress.

Continuing the presentation under the headings previously used it will be apparent how the planned progress was brought to an abrupt stop because of the absolute necessity for a concentration of naval thought on the immediate problems of the war.

UTILIZATION.

The fuel oil test plant was in a position at the end of 1916 to continue development of fuel oil equipment. In fact, until the possible value of blending was suggested, a plan had been made whereby the plant was to use a destroyer for the purpose of developing the necessary equipment to permit all vessels to handle and burn the most viscous fuel oils. This particular plan was changed to an investigation to determine the optimum percentages of blends.

Lieutenant Penn remained in charge of the plant throughout the war, but was relieved by Lieutenant J. B. Ryan, U. S. N., in January, 1918. When Lieutenant Ryan was ordered home, in December, 1918, apparently the test plant was left without a formal head until June, 1919, when Lieutenant Commander (now Commander) W. R. Purnell, U. S. N., was assigned as officer-in-charge. The testing of fuel oil, work in connection with the construction of new ships, and the training of men, occupied the time of the plant and no particular equipment advance or development could be undertaken.

In the fleet, the separate burner for making smoke screens was developed and was specified by the Bureau of Steam Engineering in 1918. Previously smoke screens had been made by by-passing fuel oil heaters or by slowing down blowers, either of which resulted in great loss of efficiency.

War experience furnished some evidence to answer the question as to what would happen when a shell hit a fuel oil tank. Numerous destroyer fires mentioned in descriptions of the battle of Jutland, were very probably fed by fuel oil. In our own experience, the *Jacob Jones* was sunk by a torpedo which hit amidships and exploded in a fuel oil tank without igniting the oil. However, when the *Shaw* was rammed and cut in two by the *Aquitania*, fuel oil from the forward tanks was ignited by sparks and set the ship on fire.

The Bureau of Steam Engineering reached the conclusion that no danger whatever was offered by the oil but that the vapor, always present in empty or partially filled tanks did constitute a

real danger. Instructions were issued, in September, 1918, to minimize the possibility of explosion of fuel oil vapors and the consequent probable fuel oil fire.

QUALITY.

Problems which are covered by this subject were the second most important class which arose during the war and which exerted a great influence over the post-war problem of fuel oil utilization.

It had been noted in the comparatively quiet days before the entry of the United States into the war that the most easily obtainable, cheapest, and most widely distributed grade of fuel oil was steadily becoming heavier and more viscous. This trend was accentuated by the commanding position occupied by Mexican crude oil as a source of fuel oil but it was just as certainly present for fuel oil produced from domestic crudes.

The magnitude of the job of modifying every oil-burning vessel by the installation of heating equipment in bunker tanks by increasing the capacity of service heaters, and possibly by increasing pump and piping capacities was so great that any simple alternative was extremely welcome. When the Bureau of Mines reported that its preliminary investigations indicated that heavy Mexican fuel oil could be made into a satisfactory fuel for use with the then present equipment by blending with gas oil, the Bureau of Steam Engineering abandoned its plan to alter equipment and decided instead to investigate the blend possibility.

It was impossible to detail a destroyer from New York or Philadelphia for the cooperative tests with the fuel oil test plant. In fact, it was rather difficult to find one anywhere which could be spared. The insistence of the Bureau of Steam Engineering and the vital importance of the test finally led, on 1 June 1917, to the assignment of the *Monaghan,* Lieutenant J. F. Cox, U. S. N., commanding, to the task which was to be carried out in Chesapeake Bay.

The Bureau of Supplies and Accounts made arrangements with the Standard Oil Company of New Jersey to furnish the fuel oil blends from its Baltimore refinery and the *Monaghan* was ordered to that point. Lieutenant Penn and his assistant, Past

Assistant Engineer William R. King, U. S. N., Retired, were ordered from the fuel oil test plant to Baltimore to assist in the tests.

Between 22 and 28 June six runs were made by the *Monaghan*. The oils used on the different runs were: (1) Navy contract fuel oil; (2) 60 per cent gas oil and 40 per cent Mexican residue; (3) 40 per cent gas oil and 60 per cent Mexican residue; (4) 30 per cent gas oil and 70 per cent Mexican residue; (5) 70 per cent gas oil and 30 per cent Mexican residue; and (6) 50 per cent gas oil and 50 per cent Mexican residue. The results of the runs as reported by Lieutenant Penn showed the blend containing up to 50 per cent of Mexican residue to be superior to the contract oil and that any of the blends tested could be burned smokelessly with but slight reduction of speed for the 70 per cent Mexican blend.

The following excerpt from a letter of Lieutenant Penn is quoted because it may very probably be applicable to any type of fuel oil which will show up in the future. The quotation is: "The main difficulty with burning Mexican oil is in convincing engineer officers that it can be done."

The *Monaghan* tests were but the beginning of the war fuel oil quality problem which is mentioned briefly in the previously cited "History of the Navy Fuel Oil Specifications." It will be more fully described here.

In preparing the schedule for fuel oil for the fiscal year 1918, Paymaster General McGowan, on "information * * * brought to the attention of S and A from such a source as to warrant serious consideration," recommended two changes in the specifications. These were to increase the minimum flash point from 150 to 175 degrees F. and to insert the paragraph:

"No residuum of any destructive distillation or so-called cracking process, be mixed or included in any fuel oil furnished to the Navy under this contract."

The Bureau of Steam Engineering adopted the second recommendation and added another one that only fuel oil from domestic crudes would be accepted. This was based on all past experience which had showed that better results in speed, steaming radius, and smokeless steaming had been obtained in using midcontinent

fuel oils than in the use of Mexican oils. These three military characteristics appeared sufficient to warrant paying a probably higher price, particularly since the Navy was at war. Supplies and Accounts objected to the restriction which would be placed on the available supply and was sustained by the Chief of Naval Operations, Admiral W. S. Benson and the Assistant Secretary of the Navy, Franklin D. Roosevelt. The latter directed that no restriction be placed on fuel oil from Mexican sources but that the Bureau of Steam Engineering draw up specifications which would insure supplying of a satisfactory oil whatever the source might be.

Steam Engineering complied by adding an upper limit of 0.75 per cent of sulphur, thus indirectly excluding large percentages of the Mexican oil. In forwarding this specification, Admiral Griffin said: "The Bureau does not consider that the present is a time to experiment with oils in the fleet, nor that a question of a small difference in cost should be permitted to interfere with the efficiency of the fleet."

Our entry into the war changed Great Britain's relationship to the American oil industry. As an ally she was now able to base her demands for fuel oil on patriotic grounds. The Committee on Petroleum of the Council of National Defence represented the oil industry. It was faced with the problem of furnishing petroleum products for all military and allied commercial needs. For fuel oil, in addition to the bulk problem of furnishing quantities sufficient for the allied navies and merchant fleets, it had to face the quality problem of meeting both the U. S. Navy and the British Admiralty Specifications.

The Committee's efforts toward solving the quality question were principally three. The first was an attempt to bring the two navies' specifications into agreement so that there would be only one grade of *Naval* fuel oil. This was never wholly successful because the British Admiralty, operating a fleet in cold northern waters, knew fairly exactly what it wanted. The British fuel oil was satisfactory for United States ships in European waters and this permitted the Committee to worry about only one grade of fuel oil for overseas shipments. The Navy at home could use a poorer grade of fuel oil.

The Committee's second step was to open up the Navy's specifications to permit poorer grades of fuel. Working on the 1918 specifications which had been in effect since 1 July 1917, the Committee, on 22 August 1917, wrote to the Paymaster General recommending deletion of both of the changes added as previously mentioned. These recommendations were approved by the Bureau of Steam Engineering and on 7 September 1917, the restriction against cracked fuel oils was removed and the sulphur limit was raised to 1.5 per cent.

Because of the importance of overseas shipments and probable transfer of fuel oil from one activity to another, the Standard Oil Company of New Jersey on 14 November 1917, recommended that the Navy change its unit of purchase from the gallon to the ton which was used by every other Navy. Supplies and Accounts canvassed sixteen oil companies on the proposed change and, although the Navy favored the change to a weight basis, only four of the companies did and the proposal was dropped. Thus an original mistake, based on unscientific commercial custom, was perpetuated.

The third step of the Committee on Petroleum was a continuation of the second. To meet the large demand for fuel oil, a majority of the oil available was too viscous and had too high a sulphur content to be acceptable to the Navy. On 6 April 1918, the Oil Division of the United States Fuel Administration entered the picture with an inquiry to Lieutenant Commander J. C. Hilton, S. C., U. S. N., Bureau of Supplies and Accounts, as to the reasons for sulphur and viscosity limitations.

The Bureau of Steam Engineering had, with the full support of the Secretary of the Navy, for the past year and a half, combated a legislative attempt to place the Navy's fuel oil specifications, tests, and purchase in the hands of the Bureau of Mines. In the letter of the Oil Division cited, it was implied, either that the Bureau of Mines could not explain the Navy's requirements or had given the impression that there was no basis for them. There was unquestionably an unfortunate lack of sympathy between the Bureau of Mines and the Navy Department.

The Engineer-in-Chief, Admiral Griffin, was adamant in his insistence that the Navy's fuel oil specifications be not lowered. The

only alternative was the one which was adopted of grading fuel oil and classifying vessels as to the grade of fuel which each could use. The four classes of ships as described by Admiral Griffin were:

1. Naval combatant vessels.
2. Vessels which could not use heavy oils but which were not required to run smokeless or to use heavy forced draft.
3. Vessels which could burn an oil of approximately 18 gravity all year around and certain others which could not handle a heavier oil in cold weather. And
4. Same as " 3 " except able to burn an oil as heavy as 16 gravity.

By executive order dated 31 July 1918, the Navy was stripped of its fuel oil specifications control by the transfer of that function to a newly created " Committee on Standardization of Petroleum Specifications " under the United States Fuel Administrator. Admiral Griffin protested this order. The protest was seconded by the Secretary but to no avail. The Navy's representative on the committee was Admiral Griffin who appointed Commander H. A. Stuart, U. S. N. as his technical adviser.

The result of the above series of events have been described in the previously cited specification history. Further comment will be in order as a portion of the final section of this history.

The Committee on Standardization issued, on 2 October 1918, its first specification which recognized the Navy's right to its standard fuel oil and provided for two additional grades, one with no sulphur limit and one with neither sulphur nor viscosity limit. The latter grade was soon thereafter split into two grades on a gravity basis and the present Bunker " A," " B," and " C " grades were born.

The war had exerted its influence and done lasting damage by the specifications expedients which circumstances dictated. The insidious results, coupled with others will rear their heads from time to time as this review proceeds.

APPLICATION.

It cannot be stated whether any thought was given to the supply of fuel oil when all of the emergency building programs were laid down. The Navy's policy to build only oil-burning vessels was naturally adhered to for the large 1916, 1917, and 1918 building

programs which, though much larger then normal, were of a routine nature. Certainly no criticism of the decision to build not only the emergency program combatant ships but also the Shipping Board's merchant ships for oil only can be based on military, operational, or engineering grounds. From a logistic standpoint there is some question and the writer suspects that this consideration was passed over rather hurriedly with the belief that "where there's a will there's a way" and that the demands of the new oil-burning ships would test the will of the Navy and the oil industry to run them. Coal, relying, as it did, heavily on the railroads for distribution ran into war difficulties which proved even more embarrassing than the fuel oil troubles so perhaps the preference for oil was wise.

The Navy's emergency building program comprised 811 vessels, divided as follows as regards fuel:

FUEL OIL: 229 destroyers, 21 sea-going tugs, 49 mine sweepers, and 60 Eagle boats.

DIESEL OIL: 20 submarines.

GASOLINE: 442 submarine chasers.

With the addition of the aviation gasoline supply, the Fuel Administration would certainly have had a Herculean task had the war continued to the commissioning and operation of this fleet.

Fortunately for the oil situation, the large majority of reserve warships and merchant ships which were operated for war purposes were coal burners. This held down, to a certain extent the advance of fuel oil toward the position of leading naval fuel.

The British alone, of the European allies was a heavy burner of fuel oil. Her fuel policy had been parallel to that of the United States as to application, for combatant vessels but such auxiliaries as she took over or built during the war were coal-burners. She could have made the situation somewhat more difficult by discarding coal entirely but, of course, her lack of domestic production of petroleum overruled even military considerations.

SOURCE.

The source of fuel oil throughout the war remained in commercial hands. Enough has been said in other sections to emphasize the importance and influence of the Mexican production.

Lieutenant Commander Landis continued his investigations in the Naval Petroleum Reserves but no formal report such as was directed by the Department was submitted. No particular change in the status of the reserves occurred except a gradually increasing knowledge of their potentialities. These details do not belong here.

Abortive attempts to obtain legislative permission to take the most valuable of the reserves away from the Navy continued. At any rate, that was the construction placed by the Navy Department on the expected effect of the various leasing bills which were presented. The oil industry of California, represented by the Western group of Congressmen wanted to exploit or develop the public lands which had been withdrawn in 1909. They preferred legislation, similar to the old placer mining laws, which would permit them to obtain ownership of the land but they were willing to accept a leasing law as the best obtainable.

The principal obstruction to the legislation was the adamant position of Secretary Daniels and the Navy Department, now absolute and immovable conservationists. The naval reserves were included in the scope of the orginal bills. Later they were specifically excepted from the general leasing provisions but were included in the saving clause which would permit the prosecution to final patent of all claims initiated prior to the Pickett Act of 1910. That meant the end of Reserve Number Two which was considered the most valuable of the three.

The Navy Department prepared a very drastic bill to accompany the leasing act. This bill, presented to the Senate by Senator Swanson, now Secretary of the Navy, contained three general provisions:

1. The Secretary of the Navy was authorized to take possession of all land within the borders of the three naval reserves, regardless of its status. As some of the land was unquestionably private property, this authorized exercise of the right of eminent domain.

2. The Secretary of the Navy was authorized to operate the reserves in any manner, either directly or by lease and to use, sell, store, exchange, or refine the oil produced.

3. Valuation of every private claim was provided for and the Secretary was authorized to reimburse the owners of the claims by

cash or by exchange of other public lands not located within the reserves.

This bill was presented and its sponsor moved that it be referred to the Naval Affairs Committee. A three day argument between Senator Swanson and Senator Smoot as leaders of the two sides followed. The point under discussion was as to whether the Naval Affairs or the Public Lands Committee had proper cognizance of the bill. The Naval Affairs Committee finally won but the bill died there.

The proposed Osage reservation Naval Petroleum Reserve was strongly advocated during the war by a number of independent geologists as well as by the Navy Department. The Secretary of the Navy requested the Secretary of the Interior to investigate the possibility of the Navy's obtaining this reserve. The answer was that the Osage Council was willing to *lease* eight scattered sections of land to the Navy Department without the payment of a bonus. These leases would have required the Navy to begin drilling and producing in the event that oil was discovered. Crude oil could probably have been produced which could have been refined or exchanged for fuel oil. Since the Navy wanted a reserve to hold for future needs and not current oil the plan was not acceptable. Legislation to permit such leasing was lacking also. No record can be found of any effort to obtain possession of a reserve in the Osage by force. There was too much opposition since the Indians wanted immediate money and the oil industry wanted the opportunity to develop and to produce oil.

When the California oil production for 1917 dropped a little below the needs of that year and it was necessary to use some of the past excess production stored above ground, pressure was brought to bear on Congress by California to throw the California reserves into full production on the grounds that oil from this source was absolutely essential to winning the war. The Navy Department threw its full force against the proposal and was helped to successful opposition by the rise of California's production to an excess over her requirements early in 1918. One of the strongest opponents to the Navy on this question was Mark Requa of California who had previously recommended operation of the reserves to forestall drainage from adjacent private lands. In 1918, Requa became head of the Oil Division of the Fuel Administration.

On 6 December 1916, two areas of oil shale lands, 41,560 acres in Colorado and 91,464 acres in Utah, were withdrawn from entry and designated as Naval Oil Shale Reserves. This remarkably rapid action, occurring only a few months after the date of the Fuel Oil Board's interim report, indicates that, of all the recommendations of that board, the one pertaining to shale reserves was the only one which neither cost the government money nor was opposed by industry individually or collectively. It was one case in which the Navy acted so far in advance of the need, that no opposition had developed. The suggestion can be read from the records that possibly the interests which hoped to develop the petroleum reserves fostered the oil shale reserve idea, hoping, once they were established, to have an argument to show that the Navy's need for the more readily usable petroleum reserves no longer existed. Several statements by various naval officials can be found in which it is insisted that the oil shale reserves in no way obviated the necessity of retaining the petroleum reserves.

It was also during the war period that first forceful proposals of a substitute for fuel oil were advanced. The one advocated was a partial substitute called "colloidal fuel" which used the liquidity of oil and the abundance and calorific value of coal in combination.

The Submarine Defence Association was organized in New York for the purpose of finding means of combating the German submarine menace. Membership was made up of shipping, insurance, and interested engineering companies and individuals. A Mr. Lindon W. Bates was chairman of the Engineering Committee of the Association. Mr. Bates had invented and later obtained patents on a colloidal fuel composed of varying percentages of pulverized coal and fuel oil held in a stable suspension by a third ingredient termed a "fixateur."

From April to July, 1918, the U. S. S. *Gem*, a converted yacht of 213 gross tons was made available to the Submarine Defence Association for testing substitute fuels which would replace oil while retaining its advantages. The object, of course, was to find means of supplying fuel to the oil burning fleets in Europe without the necessity of carrying such large quantities of oil across the Atlantic.

The *Gem* tests were not sponsored by the Navy in any way other than to make the vessel available. The results were reported as so

successful that Mr. Bates initiated an intensive campaign to get the Navy's approval of his colloidal fuel. The Assistant Secretary, to whom he wrote, turned the matter over to the Bureau of Steam Engineering which agreed to test the fuel both in the laboratory at Philadelphia and on board a destroyer of the fleet.

A requisition, dated 15 May 1919, was prepared to cover the purchase of thirty tons of colloidal fuel for delivery to the fuel oil test plant and thirty tons for delivery to the U. S. S. *Chew*, Commander J. H. Klein, U. S. N., commanding, at New York. The *Chew* had been detailed for the test by Operations at Engineering's request.

Delivery of the fuel was delayed for a time by the priority given to an order prepared for the Japanese Admiralty, then by the failure of the supplier to furnish the fixateur, and finally in September, the plant where the fuel was compounded was destroyed by fire. The *Chew* having waited in New York for four months for the fuel was required to transfer to the Pacific when the fleet was divided, the contract for the fuel was cancelled, and the test degenerated into an exchange of letters which shortly ended. The Navy never tested colloidal fuel.

SUPPLY.

The separation of this section dealing with the war period into the subjects previously adopted should show, without need for extended argument, how interrelated are the questions of source, supply, quality, and distribution. No serious problems would have arisen during the war in regard to fuel oil had not the supply question been influenced by quality restrictions and lack of distributional facilities.

Even with a somewhat reduced domestic production of crude oil, there was no shortage of crude available because the fields of Mexico were capable of off-setting the decline in the production of the United States fields. Control of Mexican production was entirely in the hands of American and British commercial interests. The fuel oil from this source would not meet either Navy's specifications. Moreover the navies were almost entirely dependent on commercial tankers for the distribution of the fuel.

Thus it was that when the bids for the Navy's supply of fuel oil for the fiscal year 1918 were opened in May, 1917, they were found to be highly unsatisfactory both as to price and conditions of delivery.

The annual contract for 1917 had carried the Navy into the war and up to 30 June 1917, at which time the Navy's contract price for fuel oil was far below the market price. In view of the unsettled condition of the fuel oil market, the Navy elected to take advantage of recent legislation which empowered the President to place demand orders for war commodities on any manufacturer or producer. Accordingly, the needs of the Navy for the three months' period, July to September, 1917, were allotted to the various refining companies in proportion to each company's share of the total production. Navy orders were placed on each company requiring it to furnish fuel oil in accordance with specifications when and in amounts needed and at a tentative price which was subject to final determination at a later date. These orders were extended from time to time throughout the duration of the war.

The oil industry was wholeheartedly in the war and spared no effort to supply every demand made by the allied combatants for petroleum products. Such difficulties as arose were handled in the most practicable manner by the Petroleum Division of the Council of National Defence. No difficulty was experienced with the Navy orders. When fuel oil was needed it was furnished if it was at all possible to do so.

The table started in Part II is here extended for the three fiscal years comprising the war period. The figures for these three years are not as accurate as the previous ones because in his annual report, the Paymaster General reported only approximate quantities used.

Year	Coal Tons	Oil Barrels	Oil Equivalent to Coal Tons	Per Cent Oil
1917	1,200,000	1,600,000	336,000	21.9
1918	4,000,000	4,600,000	966,000	19.5
1919	3,700,000	10,500,000	2,205,000	37.3

DISTRIBUTION.

It was under this subject that the greatest difficulties were met and overcome by the various agencies which were set up to handle the problem. In applying the past history of the World War to any future emergency, one fact must be kept clearly in mind. That is: The United States entered a war which was well underway on established lines. The principal demand on the United States was for bulk deliveries of men, money, and war supplies. The declaration of war, of 6 April 1917, changed the previous commercial relations between European combatants and American business to a commercial-political basis. The government of the United States was now vitally interested in the satisfaction of the customer demands of its allies.

The question of distribution during the war period cannot be restricted to the bounds set by the original definition. There was no clear cut line between the Navy and the supplier. There can be traced a certain isolation surrounding the United States Navy because matters were largely removed from military hands and placed in special civilian organizations. In one respect, a review of the records gives the Navy, and particularly the Bureau of Steam Engineering, the appearance of an obstructionist. Many proposed expedients were vigorously opposed, not in a spirit of professional jealousy but because the military mind could visualize more clearly the possible objectionable military consequences. With the civilian boards, commissions, and committees, the needs of the moment had to be satisfied in the simplest way and any interfering objections had to be brushed aside. In every case, a decision once made and an expedient adopted, the Navy threw in its full support. That and the further fact that most of the possibilities visualized by the Navy did not materialize, permitted the war to be prosecuted to conclusion without disaster in connection with the supply of fuel oil.

An excellent detailed picture of certain phases of the distribution of petroleum products during the war by Captain Paul Foley, U. S. N., was published in the Proceedings of the U. S. Naval Institute, of November, 1924. Captain Foley had been one of two members of a special petroleum mission which represented the

United States in London. The mission was interested in all petroleum products. Since the present paper is restricted to fuel oil, practically no consideration need be given to anything but the needs of the British Grand Fleet and of those oil-burning units of the Atlantic Fleet which operated in European waters.

Captain Foley states that at the end of 1917, the Grand Fleet contained 15 capital ships, 17 cruisers, and 125 destroyers entirely dependent on fuel oil. A number of other vessels were designed for either oil or coal but these ships could be utilized to nearly full advantage even if entirely deprived of fuel oil.

Of course, our entry into the war made the British Grand Fleet, officially, the force which safeguarded the United States against the German High Seas Fleet and it was vital to the United States that all necessary munitions be supplied to the Grand Fleet.

The oil-burning ships which could not use coal as well, would be practically useless hulks if the distribution of fuel oil failed. They were forced by strategic considerations to operate from a base at the head of the North Sea to which fuel oil could be carried only by tank steamer. The only tanker route was from the source in the United States, Mexico, Trinidad, or the Far East to British waters and thence around the north of Scotland. Losses of tankers in the open sea were far from negligible but the greatest losses occurred in the comparatively narrow passage the tankers had to follow at the end of their voyage. To eliminate this most dangerous portion of the route and also to shorten the voyage and increase the availability of tankers, the American mission proposed that a pipe line be built across Scotland. This plan was accepted by the British after some discussion and the pipe line was built under the direction of Mr. Forrest M. Towl of the Eureka Pipe Line Company, by a unit of pipe line workers recruited in the United States and enrolled in the U. S. Navy. The line, 36 miles long and with an hourly capacity of 100 tons, was completed about a month before the Armistice, and therefore was unable to make a very great actual contribution to the allied victory.

Before the pipe line was completed, the general situation had been greatly improved. In April, 1917, fuel oil stocks in the British Isles had decreased to 900,000 tons and were still decreasing. This in the face of the Admiralty's minimum stock requirement

of 1,500,000 tons. Partly by the requirement that all vessels crossing from America to England fill double bottoms and ballast tanks with fuel oil and by shipping fuel oil in preference even to wheat, but largely because the oil industry of the United States made the war issue paramount and, as some writers maintain, surrendered their Asiatic trade to their British competitors, the fuel oil stocks in the British Isles had been built up to 1,800,000 tons, while all current demands were also met, by the date of the Armistice.

SHORE STORAGE.

Shore storage developed during the war principally because there was a ready supply of money, much of it in lump sum appropriations. There was such tremendous public works required for the actual prosecution of the war that it was impossible to take advantage of the situation and build the storage up to the two years' supply recommended by the Fuel Oil Board and approved by the Department.

One decision, reached in 1917, was that all future fuel oil storage would be built underground and of concrete. The definite plan which was set up was not realized in its entirety, but the following summary shows what was accomplished with war money. The concrete storage, not completed until 1919, can have had no effect toward the prosecution of the war. Of course, the duration of the war was unknown until 11 November 1918, and it is very probable that the storage would have been invaluable had the war lasted longer.

Money for building storage was obtained under three types of appropriations. Each is considered separately below.

A. Money appropriated for development of major naval activities to which fuel oil storage was incidental:

Activity	Location	Capacity of Tanks Built
Engineering Experiment Station,	Annapolis, Md.	40,000 gals.
Submarine School,	New London, Conn.	480,000 gals.
Air Station,	Cape May, N. J.	200,000 gals.

Total, 720,000 gallons, or 17,100 barrels.

It is apparent that the storage built under this category was quite trivial and unimportant.

B. Money specifically appropriated for building fuel oil storage as part of the Department's plan:

Location	Number of Concrete Tanks	*Capacity* Gallons	Barrels
Melville, R. I.	2	5,138,000	122,000
Guantanamo Bay, Cuba	2	6,000,000	142,800
San Diego, Calif.	1	2,100,000	50,000
Puget Sound, Wash.	2	9,800,000	235,200
Pearl Harbor, T. H.	1	6,300,000	150,000
Total			700,000

C. Money specifically appropriated for fuel oil storage but location not disclosed. The committee hearings on this appropriation were secret and never published:

Yorktown, Va., 8 concrete tanks, capacity, 28,480,000 gallons, 680,000 barrels.

Summary of war construction:

Appropriation "A"	17,100 barrels
"B"	700,000 barrels
"C"	680,000 barrels

Total, 1,397,100 barrels or 194,500 tons.

TANKER ACQUISITIONS.

Tank vessels were obtained by the Navy during the war in four ways: Regular naval construction, lease from private owners, purchase from private owners, and assignment by the Shipping Board.

Table III of Part II of this paper lists three tankers and seven colliers with oil cargo tanks which were in commission on 31 December 1916. The total oil cargo capacity of these ten vessels was 39,602 tons. On the same date two tankers were under construction and two had been authorized and not appropriated for.

Funds for these last two were provided for by Congress on 1 July 1918. These were the last tank vessels up to the present date constructed by the Navy and for naval service.

The *Cuyama* was commissioned on 2 April 1919, and the *Brazos* on 1 October 1919. Of the other two, the *Neches* was under construction in the Boston Navy Yard on 31 December 1919, while laying of the keel of the *Pecos* awaited the launch of the *Neches*. Naval tanker construction had little effect on the war unless the shortage caused by the lack of it is considered.

One tanker named *Guthiel* was purchased and commissioned as the *Sara Thompson*. Four vessels, the *Patoka, Alameda, Ramapo,* and *Robert L. Barnes* were loaned to the Navy by the Shipping Board between 29 June 1918, and 22 October 1919.

The Navy's commissioned tanker fleet on 31 December 1919, thus consisted of ten vessels having an aggregate cargo capacity of 72,303 tons. The colliers need hardly be considered further although they rendered valuable war service.

In addition to the tankers commissioned in the regular Navy, there were 26 vessels commissioned in the Naval Overseas Transportation Service. Four of these were operated for Army account, one for Shipping Board account, and the remaining 21 were leased by the Navy from private owners and operated for Navy account. The *Joseph Cudahy,* of this service, was sunk by a submarine on 17 August 1918.

In general, the distribution of fuel oil was handled by a superagency as has been mentioned. During the war, the U. S. Navy had vessels in many European ports and on the date of the Armistice, these numbered fourteen which can be taken as illustrative of the problem.

Of the fourteen, five: Cardiff, Wales; Genoa, Italy; Murmansk, Russia; The Grand Fleet; and the North Sea Mine Force based or contained only coal-burning U. S. vessels. As a matter of fact, when Admiral Jellicoe, as First Sea Lord of the British Admiralty, requested that the United States send a division of battleships to work with the Grand Fleet, he specified that they should be coal-burners because of the inadequacy of the fuel oil supply at the northern bases. At Corfu and Plymouth, England, only gasoline-burning submarine chasers were based. Queenstown, Ireland;

Gibraltar; Southampton and Liverpool, England, based oil-burning vessels but the distribution was assigned to the British.

There were thus left only three bases in Europe from which the *distribution* of fuel oil was specifically charged to the U. S. Navy. It must be remembered, however, that the U. S. Navy was largely responsible for the *supply* of fuel oil to many of the other bases. The three bases were Berehaven, England; Brest, France; and the Azores.

The force based on Berehaven was primarily a division of heavy ships, so placed as to be available for convoy protection against a possible German battle cruiser raid. The battleships were the *Utah, Nevada,* and *Oklahoma,* the last two of which were oil burners. They were adequately cared for by maintaining a loaded tanker as needed at the base.

The advance base at the Azores was merely a stepping stone to Europe. It was used for a repair and refueling station for ships crossing to Europe. Those of the smaller vessels whose fuel capacity was insufficient for the crossing were routed via the Azores. The supply of fuel oil was maintained by keeping a loaded tanker at the base.

Prior to the establishment of the Azores base it had been necessary to maintain floating bases in midocean for refueling. Ships crossing were required to rendezvous at sea with a tanker which kept a moving position until her tanks were emptied. Fortunately the destroyers of the Atlantic Fleet had perfected methods of oiling at sea during the 1917 winter maneuvers in southern waters and were able to refuel without difficulty. All of our first destroyers which crossed were refueled in this way.

In France, it was necessary for the United States to provide shore storage for both naval and merchant vessels. A total storage of 2,000,000 gallons was constructed at La Pallice, Brest, and Le Havre. This was all disposed of by sale to the French government after the war.

NAVAL FUEL OIL BOARD.

The writer considers that the recommendations of the Fuel Oil Board, as summarized in Part II of this history, were basic and that they indicated the proper course for the Navy to follow in

regard to the fuel oil problem. For this reason, it would be well to summarize the progress made, during the war period, toward realization of the board's recommended goal.

a. Establishing clear title to the existing reserves.—Conditions in regard to the title of the reserves were no better in 1919 than in 1916. The Navy had been successful, however, in preventing any legislation which would make the Navy's title any less secure.

b. Acquisition of an Osage reserve.—Efforts along this line were unsuccessful, due apparently to the failure of the Interior Department to understand exactly what was wanted. Had the necessity arisen the Navy could have produced oil from the Osage reservation under lease for current use during the war. The offer of the Indians to lease for current production and their disregard of the general proposal of the Navy Department, brought this recommendation to an unsuccessful conclusion. As far as can be determined from the records the question has never been reopened.

c. Acquisition of oil shale reserves.—This recommendation was realized 100 per cent.

d. Construction of a Navy refinery.—This recommendation does not appear to have been very strongly supported. The bill drafted by the Department to remove the petroleum reserves from consideration under the proposed leasing bill, authorized the secretary to *refine* oil from the reserves. No record can be found of any attempt to obtain authorization from Congress for the construction of the proposed refinery.

e. Construction of shore storage.—At the end of 1919, slightly less than 2,500,000 barrels of storage had been provided for. This was approximately three times the 1916, one and one-half times the 1917, half the 1918, and one-fourth of the 1919 consumption. Thus in 1919 the storage available was far below the recommendation of the board.

f. Construction of tankers.—The board made no detailed recommendation as to the tanker capacity needed. Due to the war demands, the capacity of the Navy's tankers had about doubled but the total capacity was equal to only about eighteen days' consumption at the 1919 rate.

g. Investigation of methods of reducing viscosity.—Tests of blends had been successful but they covered only Mexican heavy crude and midcontinent gas oil. The investigation was entirely adequate for the time and sufficient for war needs. The basic value was limited because of the restricted types of oil considered.

h. Investigation of the actual value of the reserves.—This investigation was under way but it had been discovered that the value of the reserves could not be determined as easily and hurriedly as the Department had desired. Progress of the investigation can be considered as satisfactory.

i. Establishment of an "Oil Office."—Nothing was accomplished on this recommendation and the records do not show that anything was attempted.

j. Preparation of specifications for fuel oil.—This question was handled satisfactorily for war purposes but the work done, influenced as it was by the special circumstances of the war, cannot be considered as of a permanent character.

The above summary indicates how expediency, rather than planned development, ruled during the war and also what should have been the course of events during the post-war period.

THE NAVY'S SOLDERED FITTINGS.

By MASON S. NOYES, MEMBER.*

About a year prior to the time when this is being written the Bureau of Engineering of the Navy Department decided that the use of non-ferrous piping and pipe fittings designed to be soldered together had progressed far enough industrially to be utilized on naval vessels of all types. It was evident that this "new" pipe joint offered certain marked advantages to naval construction. American standard pipe walls are doubly thick—the first portion being selected to withstand internal pressure and corrosion, and the second portion to permit threads of proper depth and to provide somewhat for external corrosion. By substituting a soldered joint for a threaded one, the second portion of the standard pipe wall could be left off and a substantial saving made in the weight allowance for piping. The soldered joint itself had proved to be adequately strong, light, and reliable in the hands of average workmen. Poor thread cutting was eliminated, with its attendant losses in short lengths of pipe, time, and labor. Preparation of the pipe ends was simplified and the time shortened. Bench assembly to exact dimensions was secured. Bends were more easily made. True stream-line flow was provided, the familiar "rushing" sound of liquids in pipes being thus almost entirely removed, together with the oft-noticed erosion of pipe or fittings near abrupt changes in internal diameter. It was obvious, however, that workmen installing piping on board naval vessels would have to learn how to make the new joint. Since similar workmen on land installations had readily become proficient there appeared to be no reason why shipyard pipe-fitters could not do equally as well. Thereupon the Design Division of the Bureau of Engineering undertook the development of the necessary specifications and standards for the use of these fittings on naval vessels.

* Design Division, Bureau of Engineering.

At the same time the Bureau of Construction and Repair, having likewise considered the possibilities of the soldered joint for ship's piping under its cognizance, had prepared specifications for ferrous (corrosion resisting steel) tubing, fittings, and valves which were deemed suitable for the pressures and installation conditions peculiar to hull piping. One major difference marked this parallel program from that undertaken by the Bureau of Engineering. Where the latter proposed to restrict its means of joining to soldering, the Bureau of Construction and Repair resolved to use welding for most of its work and to retain soldering strictly as an aid in assembly and in subsequent alterations. This did not represent a sharp division of opinion between the Bureaus but merely indicated once more the difference between the piping conditions met by each. In the interval since then the development has resulted in the two methods drifting somewhat closer together.

The first move by the Bureau of Engineering was to test at the Experiment Station the possibilities of commercial screwed brass unions. The pipe-threaded portions were bored out for inserting the tubing, the innermost thread being left as a shoulder to meet the end of the tube. These unions provided insufficient depth at the bored-out part for a soft solder joint and they proved to vary in dimensions and to lack an unvarying inside diameter. Two kinds of high-melting-point (brazing) solders and one soft solder were used and much valuable information on joint clearances, strengths, fluxes, and heating was obtained. Particular attention was paid to the effect of brazing temperatures on fitting materials containing a relatively large percentage of lead.

One particular method of test may prove interesting. Specimens consisting of two lengths of copper tubing connected by a brass union and brazing solder were arranged for test as shown in Figure 1. Steam at 100 pounds pressure and 425 degrees F. was admitted to the specimens. After the tubes and fittings had attained the maximum temperature, the steam was by-passed and cold water was admitted to the specimens, until the tubes and fittings had dropped to the temperature of the water. This sudden change in the temperature combined with the load suspended from the midsection of the specimen subjected the tubing, fitting, and joint to two sudden severe shocks during each cycle. This cycle was

repeated 100 times. Following these tests, the pressure was increased to 200 pounds gauge at 425 degrees F. and the tests continued for 100 cycles. During the course of these experiments note was made of any joint leaks or weeping of the fittings. The specimens were arranged across knife edges, spaced as shown, and were loaded across the fitting so as to produce a maximum fiber stress in the tubing itself slightly in excess of the proof stress. This load was sufficient to cause the specimens to sag somewhat during the course of the test but not sufficient to result in excessive bending. No leaks developed at the brazed joints although all tubing showed considerable deformation at the conclusion of the tests. Moreover, no unions revealed any weeping although they contained at times as much as 7½ per cent of lead.

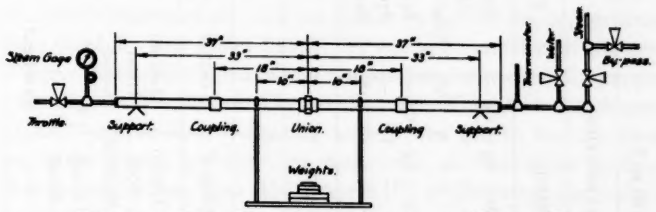


FIGURE 1.—LINE DIAGRAM OF STEAM AND WATER TEST.

At this same time a proposition utilizing commercial unions and fittings having a ring of brazing solder inserted into the wall of the fitting in way of the copper tubing was investigated and rejected in favor of a joint adapted for either a brazing or a soft solder.

The first move having established the impracticability of using commercial screwed unions and fittings, the Bureau of Engineering proceeded to bring out its own design of soldered fittings. In order to do so a choice of solders had to be made and the next test results of the Experiment Station gave information on solders. These divided naturally into the two types mentioned above—high-melting-point (" brazing " or " hard ") solders and low-melting-point (" soft ") solders. The data in Table I were obtained:

The development of the last " hard " solder listed was completed after the Bureau's designs for solder fittings had been finished. It is a more expensive material than the first " copper-silver " solder.

TABLE I.

CHARACTERISTICS.

Name	Type	Composition	Flux	Melting point	Flow point
Tin-Lead (50:50)	Soft	Sn50 Pb50 Sb— Ag— Cd—	$ZnCl_2$	360 degrees F.	415 degrees F.
Tin-Antimony (95:5)	Soft	95 — 5 — —	$ZnCl_2$	450 do.	465 do.
Cadmium-Silver (95:5)	Soft	— — — 5 95	$NH_4Cl,ZNCl_2$	640 do.	740 do.
Copper-Phosphorus (P.C.)	Hard	Cu90-95 P5-10 Zn— Ag— Cd—	Special	1290 do.	1380 do.
Copper-Silver (SF)	Hard	80 5 — 15 —	Special	1200 do.	1300 do.
Copper-Silver (EF)	Hard	15.5 — 16.5 50 18	Special	1185 do.	1220 do.

TESTS.

	Tension			Flow			Steam and Water
Name	Temp. 75 deg. F.	212 deg. F.	425 deg. F.	75 deg. F.	212 deg. F.	425 deg. F.	425 deg. F.
Tin-Lead	Good	Poor	Liquid	Medium	Bad	Liquid	None made
Tin-Antimony	Good	Fair	Poor	Good	Medium	Bad	None made
Cadmium-Silver	Medium	Medium	Medium	Good	Good	Fair	Good
Copper-Phosphorus	Excellent		Good		Excellent		None made
Copper-Silver	Excellent		Good		Excellent		Excellent
Copper-Silver	Excellent	——	——	Excellent	——	——	None made

With the above solder data the Bureau of Engineering rapidly completed its own design of soldered fittings, as shown on its standard drawings 3-S-524, 3-S-525, and 3-S-526. Since the use of the pipe thread was to be minimized, union-end globe and gate valves based on standard composition screwed valves and fitted with soldered union connections had to be developed, as shown on Bureau standards B-164 and B-165. Also in order to allow standard composition flanged reducing fittings to be used as usual on the larger lines, flanged fittings combining a standard solder recess for the tube and a standard flange for the flanged reducing fitting were laid out, as shown on Bureau standard B-168.

Up to this time copper or brass pipe and tubing had been specified for lines carrying steam under a vacuum or for steam-heating, and for lines carrying lubricating oil, low pressure fresh water, low pressure compressed air, and sea water (provided the copper lines were carefully soft solder-lined). It was apparent that, if the new design could meet the maximum temperature restriction for copper (425 degrees F.), it would be satisfactory. Therefore the cheaper of the "copper-silver" brazing solders was chosen for the more particular piping systems. As both fresh and sea water services operate under less than 150 degrees F. temperature the Navy's standard tin-lead solder was chosen, its creep characteristics at such temperature being satisfactory.

The lengths of joints in the final design of the Navy's soldered fittings have been based on its tin-lead solder but the method of soldering and the location of the hole and groove are determined by the "copper-silver" brazing solder. The provision of union-end fittings became a *primary* requirement, for the purposes of decreasing the number of separate unions and fittings required, and of saving both the amount of space wasted thereby and an excessive number of joints.

When commercial unions were being investigated by the Experiment Station, manufacturers of soldered fittings already on the market were invited to submit samples of their fittings, solders and fluxes, and to supply whatever information they might wish for the Navy's use. Examination of each maker's commercial designs revealed the following points of disapproval: (*a*) all but one were based on a "nominal" tube size (like $\frac{5}{8}$, $\frac{7}{8}$, $1\frac{5}{8}$) instead of the

"I.P.S." tube size which is the Navy's standard; (*b*) all had very short radii at ells and tees with the result that the soldered joints were too near each other, especially for a copper-silver solder; (*c*) several lacked a shoulder inside the recess against which the tube might butt, thus lacking stream-lining—in one case the tube could enter the run sufficiently to block the branch outlet; (*d*) all entirely lacked union-end fittings, flanges, and, in some cases, satisfactory and rugged unions; and (*e*) several had patented or proprietary features. However, it was discovered that one of the best of these patents had been developed at the Portsmouth Navy Yard to such an extent that in the subsequent settlement of patent interferences the Navy had retained the unquestionable right to make or have made for Naval use the patented features.

Therefore this protected feature—the hole-and-groove—became an essential detail of the Navy design. At the same time all the above points were provided for in the Navy line: I. P. S. tube sizes and minimum wall thicknesses, with joint details designed to match; generous radii at tees and ells with resultant adequate separation of the joints; true stream-lining throughout; husky unions and a complete line of union-end fittings, union-end valves, and flanged fittings; and no patent interference. See Figure 2. It is believed that no other line of soldered fittings is so complete for general piping purposes.

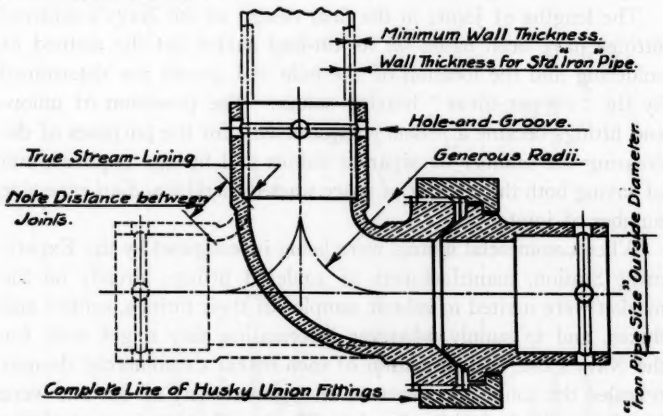


FIGURE 2.—CROSS-SECTION OF 90-DEGREE UNION ELL, UNDIMENSIONED POINTS OF ADVANTAGE INDICATED.

The following table gives selected data from the various tests made on soldered fittings:

TABLE II.

HYDROSTATIC TESTS.

Fitting	Solder	Tubing Thickness	Max. Pressure Gauge lbs./sq. in.	Failure in
		inch		
¾ inch Tee	Copper-Silver	.114	4500	Tee
1¼ inch Tee	Copper-Silver	.065	2500	Tubing
1½ inch Union	Copper-Silver	.095	3000	None
¾ inch Union	Copper-Silver	not recorded	2200	Ground joint
¾ inch Tee	Copper-Silver	not recorded	3000	None
90 degree Ell	Copper-Silver	not recorded	3000	None

TENSILE STRENGTH (TOTAL POUNDS PULL).

		Tin-lead Solder	Copper-Silver Solder			Tin-lead Solder	Copper-Silver Solder
I. P. Size	Tubing Strength	Broke in Joint	Broke in Tubing	I. P. Size	Tubing Strength	Broke in Joint	Broke in Tubing
¼	6050	6100 ave.	6000	¼	8325*	—	4950*
¾	13500	11800 ave.†	12500	¾	17150*	12700 ave.	10400*
1¼	22100	20675 ave.	18900	1¼	27050*	—	17150*
¾	13500	12800 ave.†	—	—	—	—	—

* Tubing was very hard-drawn, causing abnormal drop in strength when annealed by brazing solder heat.
† Maximum value 13400 pounds.

At the very outset specifications were set up for seamless copper tubing, under SGS (48)-39, for conveying steam, lubricating oil, compressed air, and fresh water. Information was sought that would allow the abandonment of solder-lined copper tubing for sea-water. Current research had not yet reached a point that gave an assured answer. Nevertheless the copper-nickel (70 Cu-30 Ni) alloys were selected by the Bureau of Engineering as having the most promise. Following the design of the fittings, specifications for copper-nickel alloy seamless tubing were prepared, checked, and issued for future use on sea-water piping. Since then steps have been taken to submit this tubing to actual service conditions aboard ship.

In choosing a reliable composition for the fittings, the Bureau of Engineering was faced with a heating problem similar to that of its brazing flanges where composition " F "—a low brass having a melting point of only 1830 degrees F.—is raised to a temperature hot enough to suit a brazing spelter melting at about 1600 degrees F. However, as stated above in connection with solders, the " copper-silver " brazing solder that was selected had a flow point of only 1300 degrees F. and therefore a more easily cast material than composition " F " could be chosen. The naval bronze called composition " G " has a melting point of about 1950 degrees F. and a long and excellent reputation for general use aboard naval vessels. Modified with a very small amount of lead for better casting and machining qualities, it is known as composition " M." In view of the amount of careful machining on and the relatively thin sections of the new solder fittings, composition " M " was chosen. Several of the various kinds of smaller fittings could be machined or forged from bar stock and allowance was made for these to be of rolled naval brass (composition " N-r ") if desired.

The Bureau of Engineering has been—and still is—urged to allow a cast composition, known to the foundry as " 85 and 3–5's " and to the Navy as " ounce metal," in place of composition " M." Though this alloy appears to be used extensively ashore, the Bureau of Engineering has no qualitative experience with it afloat—especially after it has been brazed—and so far has seen no adequate reason to abandon its own reliable alloy.

At present the program involving the Navy's soldered fittings as applied to pipe systems under the cognizance of the Bureau of Engineering may be summed up as follows:

(*a*) *Services and Solders.* Low pressure steam, compressed air, and lubricating oil:—"copper-silver" brazing solder required throughout. Fresh water and sea water:—Tin-lead solder throughout.

(*b*) *Temperatures and Pressure.* 425 degrees F. maximum for "copper-silver" brazing solder joints; 150 degrees F. maximum for tin-lead solder joints; 200 pounds per square inch, gauge, as a maximum throughout for "copper-silver" brazing solder joints, and 100 pounds per square inch for tin-lead solder joints.

(*c*) *Tubing.* Copper, seamless, for all services except sea-water. Copper-nickel alloy, seamless, for sea-water.

(*d*) *Fittings.* Composition M or composition N-r. Union and all-solder styles. Hole-and-groove (Navy) type of joint.

(*e*) *Intent.* To supersede all brass and copper pipe, and brass tubing. To supersede all pipe-threaded joints, in so far as possible.

This program is not yet complete. When the standard drawings are finished, there will be a complete line of all-solder fittings from ½-inch I.P.S. to 5-inch I.P.S. inclusive, and of union-end fittings and valves from ½-inch I.P.S. to 2-inch I.P.S. inclusive. In lieu of union-end fittings and valves in sizes above 2-inch I.P.S., it is intended to develop a line of soldered-hub flanges suitable for soldering to tubing and to use standard flanged valves. Tentative dimensions for soldered-hub flanges have been selected and tests are now being made at the Experiment Station. Flange test results so far obtained indicate that a large saving in time and material is possible, that the strength of the joint is increased, and that the jointing is uniform entirely around the circumference. Since the heating required is less, and since this flange has a deep soldered hub with a closer fit on the tube, it is hoped to eliminate refacing of the flanges. The warpage of the flange appears to be almost negligible and the close fit of flange hub and tube makes possible an accurate squaring up of flange face and tube axis.

METHODS OF USE.

What has already been stated is very briefly a history of the development of the Navy's soldered fittings and an outline of the place they are designed to take in the piping for which the Bureau of Engineering is responsible. All this is of little practical value to the man who is to lay out the piping or to the man who is to install it later. Therefore it is now proposed to set down a few facts concerning tube materials, solders, and fluxes; to reproduce the instructions issued by the Bureau of Engineering for making satisfactory soldered joints in its fittings; to comment briefly on some of the reasons behind certain instructions; and to suggest ways and means for the economical lay-out of pipe lines which use these fittings.

Repeating in outline form the Bureau's service specifications, we note that they fall easily into three groups:

Service.	*Tubing.*	*Solder.*
Steam (vacuum and l.p.), Lub. Oil, Air (l.p.)	Copper	Copper-Silver
Fresh water (l.p.)	Copper	Tin-Lead
Sea water	Copper-Nickel Alloy	Tin-Lead

The fittings, of course, are always to be of naval bronze, Composition M, or rolled naval brass, Composition N-r.

All of the materials listed in the preceding paragraph have some influence on the type of flux used. For the flux, the first item which must be considered is the type of solder—"soft" or "hard." Later, if service conditions require, the particular composition of solder best suited for the service conditions may enter the picture. The second item which must be considered concerns the characteristics of the particular objects to be joined. In this case it is the tubing and fitting material.

The flux has three "jobs" to do during the act of soldering. It must *clean* the surfaces to be soldered; it must *protect* these surfaces from further oxidation; and it must give an unmistakable *signal* that these surfaces are at or near the correct soldering temperature. In a paper presented at the 1934 fall meeting of the

American Welding Society, Robert H. Leach, an authority on silver solders, states these requirements slightly differently:

"The function of the flux is three-fold: it offers a protective coating which tends to prevent oxidation, dissolves any oxides which may form, and assists in the free-flowing of the solder.

"The last-named function is *important** and it has been found advantageous in some cases to use flux in furnaces having a reducing atmosphere, such as hydrogen, where oxidation could not take place.

* * *

"Fluxes that fume badly are, of course, objectionable to the workman but the flux must be *fluid* and *active* at the temperature used for soldering.

* * *

"It cannot be too strongly emphasized that *clean joints* and careful attention to the selection and *use* of a suitable flux are essential to the production of sound joints."

The selection of the proper flux has received as careful attention as have the solder and the piping materials. The Navy's existing specifications for tin-lead solder and for paste soldering flux were already matched and tested in service. However, this particular flux is not adapted to a resistant material like the copper-nickel alloy. Consequently the Experiment Station has undertaken the making of an approved list, for fluxes suitable for tin-lead solder and copper-nickel alloy tubing, from among capable manufacturers marketing such a flux. Happily the development of the two copper-silver solders was paralleled by that of an excellent flux designed especially for silver solders and for base metals even more resistant than copper-nickel alloy. It can be seen from this that, as indicated in the first paragraph above on fluxes, both the type of solder and the metals to be joined have dictated the choice of flux specified by the Bureau of Engineering.

The following instructions have been prepared from the experience gained by all who had a part in making the tests of joints at the Experiment Station. These included certain final tests made for the sole purpose of settling minor details of fittings and of

* The italics are the writer's, not Mr. Leach's.

the soldering methods. The instructions, as the Bureau's Standard B-172, have been issued to all naval activities concerned (and via them to all private shipyards now building Naval vessels) for the express purpose of setting up a definite and preferred method for making soldered joints in Navy soldered tube fittings. It is hoped that these instructions can aid those new to the soldered tube joint to make satisfactory assemblies from the start and can prevent those already more or less familiar with it from making any minor mistakes, or possibly major ones resulting in added expense and lost time.

PREPARATION AND ASSEMBLY OF THE JOINT.

1. Cut the tubing ends square with a fine hack saw (32-tooth saw recommended). The saw must be suitably guided to assure a square cut. Remove outside and inside burrs.

2. Using *sandpaper* or *emery paper,* clean the outside of the tubing at the end equal (at the least) to the depth of the recess of the fitting. Remove all dark spots.

3. Using *sandpaper* or *emery paper,* clean the inside of the recess of the fitting where the joint is to be made, until the surface becomes bright, thus removing any foreign substance.

4. Scribe a line around the surface of the tubing at a distance from the end of the tubing of one inch plus the depth of the recess.

5. Apply a coat of soldering flux to the outside of the tubing and the inside of the recess with a small brush, care being taken to apply flux evenly but not too thickly. See paragraphs 7 and 12.

6. Slip the tubing into the fitting until it will go no farther and turn the tubing back and forth once or twice to insure an equal distribution of the flux. Check the scribed line, for the 1-inch clearance beyond the mouth of the recess, to assure that the tubing has fully entered the recess.

SOLDERING WITH SOFT (TIN-LEAD) SOLDER.

7. Use " 50:50 " Tin-Lead Solder conforming to Grade A, Navy Department Specifications 46S14. Apply Soldering Flux (paste), Navy Department Specifications 51F1, on Copper tubing joints; and a suitable approved flux on Copper-Nickel alloy tubing joints.

8. Using a blow torch, apply the flame around the fitting until it is evenly heated. Melt a drop of solder into the solder feed hole. Apply the flame to the fitting near the feed hole, until the drop of solder disappears, indicating that the proper temperature has been reached.

9. For the smaller fittings having one feed hole, apply the flame to the side opposite the feed hole. Feed the solder through the feed hole, until a complete ring of solder appears at the mouth of the recess. For the larger fittings having one feed hole, apply the flame to the fitting near one side of the feed hole until the drop of solder melts; then, continually sweeping the flame around the fitting toward the side opposite the feed hole, feed the solder until the ring appears at the mouth of the recess. Repeat this operation on the other side of the feed hole. Seal the solder feed hole and remove any surplus with a small piece of cloth.

10. For the fittings having four feed holes, start to heat the fitting near the uppermost hole. Heating always between the holes, proceed as described in paragraph 8 working around toward the lowest hole and sealing *it* the last of all. Remove any surplus solder.

11. *Important.*—Permit no strain on the soldered joint until it is cool.

SOLDERING WITH SILVER SOLDER.

12. Use strips, size 0.050 inch × ⅛ inch, of Grade III Silver Solder, Navy Department Specifications 47S13. Apply Silver Brazing Flux, tentative Navy Department Specifications 51F4.

13. Use an oxy-acetylene blow torch, with slightly *reducing* flame. For tubing under ¾ inch I.P.S., a Number 5 tip is preferable. For tubing ¾ inch I.P.S. and up, a Number 8 tip is recommended.

14. Heat the entire surface of the *fitting* until the flux becomes liquid, as revealed around the mouth of the recess. Next, manipulate the torch so as to raise the temperature of the fitting in the vicinity of the feed hole. Feed the solder into the hole. Manipulate the torch with a sweeping motion so as to draw the solder away from the hole. Continue this process until a complete white solder ring is seen around the mouth of the recess. Seal the solder feed hole.

15. For the fittings having one feed hole, direct the flame toward the side opposite the feed hole at a spot near and above the hole, using the sweeping motion of the flame to draw the solder to the side opposite the hole. Repeat this operation on the other side of the hole when it is necessary to complete the white ring.

16. For the fittings having four feed holes, start to heat the fitting near (and above) the uppermost hole. Heating always between the holes, proceed as described in paragraph 15, working around toward the lowest hole and sealing *it* the last of all.

17. *Important.*—Permit no strain on the soldered joint until it is cool.

18. Wire-brush the pipe and fitting and wash the complete assembly so as to remove all excess scale and flux. Hot water is preferable.

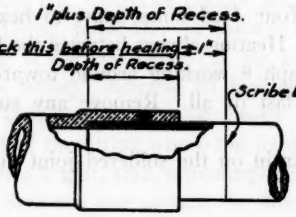


FIGURE 3.—SHOWING SCRIBE LINE.

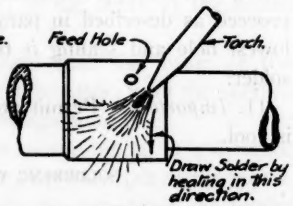


FIGURE 4.—SHOWING HOLE AND FLAME.

COMMENT ON SOLDERING INSTRUCTIONS.

Note paragraph (1) of the instructions. The shoulder at the bottom of the recess in the fitting is machined square with the axis and the tubing can be cut truly square by any of several simple devices on the market for guiding the hack-saw. Therefore there is no reason for a tube to be so cut as to produce a gap between tube-end and shoulder.

It is intended by paragraphs (2) and (3) to prohibit the use of a file or any other heavy—or rough—cutting means which may remove too much of the thin tube wall or interfere with the very necessary small clearance between tube and wall fitting.

The use of the scribe line of paragraphs (4) and (6) is the easiest way for the pipe-fitter to prove before the soldering is begun that his tubing has entered the fittings as far as it can possibly go. In assembling a section of piping it is usual for the tubing to work partly out of the fittings at one or more points. Soapstone or sharpened chalk is more satisfactory than lead pencil.

The warning in paragraph (5) as to spreading flux " evenly and not too thickly " is included more because extra flux does no beneficial cleaning and does seriously interfere with the free flow of the solder than because economy is sought.

By heating up the entire joint as the first act in soldering (paragraphs 8 and 14), the fitting and its tubing together are allowed to expand evenly and to soak up heat until the joint has almost reached the melting point of the flux. In fact, " beading " of any excess of the silver solder flux is the signal for the next heating operation. For the soft solder, the writer prefers the drop of solder as a heating signal. This indicator is not needed for the silver solder as the " beading " of the flux is more visible and as the action of the flame played directly on the silver solder spoils the latter and clogs up the feed hole with it. When the flame is used to heat the fitting *near* the feed hole, capillarity is set up for the solder and the fitting itself is more easily and quickly raised to the final (flow point) temperature. As given in Table I each solder has a short temperature period during which it is molten but does not flow. It must have reached the flow point before it can " run " into the joint via the feed hole. This means that the flame must raise the fitting (and its adjacent tubing) to a temperature somewhat higher than the flow point of the solder.

Exactly how the final heating instructions in paragraphs (9), (10), (15), and (16) are to be applied depends upon the position of the joint—horizontal, vertical, or inclined. These paragraphs have been written as if the joint is in the horizontal. Therefore it is suggested that each workman set up his own procedure for making soldered joints in various positions, starting with practice assemblies of soft solder joints followed by silver solder in others as skill and heating knowledge increase, and using the above paragraphs only as a general guide. In any case the joint is not complete until the white solder ring shows entirely around the tubing

at the fitting and, following this, until all solder holes have been filled up ("sealed") with solder.

Unless the newly soldered joint has cooled until it has begun to "set" at nearly its maximum working strength we are obviously unwise to allow the joint to take any pull or other strain (see paragraphs 11 and 17).

Some of the ingredients in the silver brazing flux are not wanted in either feed water or drinking water though the quantity is minute. This flux is also hydroscopic, *i.e.*, it absorbs water, causing the usual green discoloration on the copper tubing. Therefore paragraph (18) should be thoroughly carried out. This flux readily dissolves in water and can be thinned easily to any desired consistency.

There is one point in heating that is not included in the instructions. It applies particularly to the copper tubing because of its high heat conductivity. In heating a joint hot enough for silver solder where the piece of tubing is relatively long and especially for tubing of the largest sizes, it is necessary to heat the tubing briefly from time to time near the fitting. This assures that the tubing does not get too cool for a flow of solder, even though the fitting itself may be adequately heated. A cool tube "chills" the solder, with resulting failure to penetrate and fill the joint.

Finally, an outline of what the Bureau of Engineering considers good piping lay-out for soldered tubing and fittings may prove of value to the piping draftsman and any others concerned. *First,* let it be said that the pipe thread is to be discarded wherever possible. However, a certain number of pipe taps cannot yet be avoided and for such connections threaded union ends have been provided.

Second, these same pipe taps are based on the use of standard screwed pipe and not on tubing. Therefore it is expected that in most cases—particularly for sizes ¾-inch and smaller—a tubing one size smaller than the pipe tap size can be used, because the internal diameter of the tubing is larger than the inside diameter of the pipe of the same "Iron Pipe Size."

Third, every section or branch of pipe should begin and end with a union or a union-end fitting. Such is the intent of the General Specifications for Machinery applying to piping details. It is obvious that with pipe-thread joints replaced by permanently sol-

dered joints, some such means must be provided for easy assembly by the pipe-fitters. The Bureau of Engineering, on the other hand, is interested first of all, in requiring an ample number of union joints, to secure sufficient accessibility and flexibility for maintenance purposes. Accessibility mainly means that, where examination, overhaul, or dismantling requires removal of some soldered piping, there are union joints conveniently located in the piping for such removal. Flexibility, however, means that where repairs, alterations, or additions must be made to the piping section itself, there are union joints suitably located to expedite the work and to minimize the amount of piping that is affected thereby.

Fourth, it is also the purpose of the Bureau of Engineering to encourage the assembly *at the work bench* of the piping section between any two union joints because of the facility with which the soldered joints may be made at the work bench and because of the ease with which the completed section can be set in place on the ship by the use of the union connections.

Fifth, the use of reducing fittings is being actively discouraged. Standard reducers have been provided so that any outlet of a standard fitting can be reduced in tube size as desired. It is accepted that this combination of two separate fittings requires an extra joint and is rather long—too much so, in fact, for cramped locations. The extra joint actually has no bearing on the matter as the soldered joint is, at least, equally as strong as the tubing or fitting. The extra length can be entirely taken care of, it is believed, in the lay-out of the piping—especially in avoiding any cramped installations. The chief concern here is in connection with the replacement of special reducing fittings originally installed by the shipbuilder. It is not desired to stock at, or to encourage the use of reducing fittings, by the Navy Yards (or by any fleet repair forces), as stocking involves too many difficulties and expense. Likewise, purchase of scattered sizes of special reducing fittings for replacement purposes may easily cause serious delay in fleet maintenance work at Navy Yards.

Sixth, as stated in the first part of this article, the globe and gate valves to be used in soldered pipe lines have been based on the standard composition screwed valves as made commercially at present by a number of manufacturers. This involved only a slight change at the inlet and outlet to provide some extra metal

for the external male threads which match the standard soldered union ends. These valves thus provide additional union joints for the assembly of the piping and insure that in case of damage to or failure of any valve another one can be readily installed in its place. This is one distinct advantage which the union-end valve has over the usual screwed valve. These valves are shown on Bureau standards B-164 and B-165.

Seventh, and last, piping that is too large for soldered fittings—or for soldered unions, in some cases—have flanged joints and flanged fittings. Where a branch is of a reduced size for which soldered fittings have been provided, a "flanged fitting" as described in the first part of this article has been designed. Though the tube is permanently soldered to the "flanged fitting," the joint made up at the flange itself is an adequate "union" connection.

It is believed that the above points cover the salient advantages and aims of the Bureau of Engineering in the proper lay-out of piping involving the use of soldered fittings. It is acknowledged that many minor points have not been mentioned and may arise later, but for the most part it is anticipated that these will be taken care of by the ingenuity of the shipbuilders themselves just as other minor complexities in screwed and flanged piping have been met in the past.

Before leaving this subject acknowledgment is made of the interest and information received from various companies manufacturing some form of soldered tube fitting or solder and its flux. Particularly are thanks extended to several companies who have generously given materials and the active aid of their most experienced representatives, and to those of the staffs of the Bureau of Engineering, the Bureau of Aeronautics, and the Naval Engineering Experiment Station who have contributed so much of encouragement and data toward the initial progress of these fittings.

Unless experience from land installations is entirely misleading, it is believed that a steadily increasing satisfaction will be derived by designers and pipe-fitters in the various shipyards at present building Naval vessels as their familiarity with and use of the Navy's soldered fittings accumulates.

Much, too, is expected of the development of the Navy soldered-hub flange when completed, and the new program for the use of silver soldered steel tubing and fittings for low pressure, high temperature steam service.

RESULTS FROM THE CHANGE IN THE NAVY'S METHOD OF BOILER-WATER TREATMENT.*

By Robert C. Adams. Jr., Member.†

Between June, 1931, and August, 1933, the United States Naval Engineering Experiment Station was engaged upon an investigation of methods and means of boiler feed-water treatment. The object was to develop or to discover the method best suited to Naval use which would provide scale-free evaporative surfaces, corrosion prevention, and desirable steaming conditions. A discussion of the methods and the results of the investigation and of the method of feed-water treatment adopted as a result was presented before the annual meeting of the American Society of Mechanical Engineers in December, 1933.‡

The necessary treating chemicals were combined into the form of a single compound, for the sake of simplicity. The formula of Navy Boiler Compound 1933 is forty-seven per cent anhydrous disodium phosphate, forty-four per cent soda ash and nine per cent corn starch. These proportions were chosen after extensive tests in small experimental boilers had shown the mixture to be satisfactory under the most severe feed-water conditions disclosed by a survey of conditions throughout the Fleet.

The survey comprised analyses of samples of water from each of twenty ships of the Fleet. Since twenty ships are less than five per cent of the Fleet, it was possible that in even a large number of ships worse water conditions existed than were indicated by the survey. In order to make sure that the new methods would prove themselves completely satisfactory, the Bureau of Engineering arranged to have quarterly reports forwarded to the Engineering Experiment Station from over fifty selected vessels. Table I shows

* This paper has been revised from one with the same title presented at the annual meeting of the American Society of Mechanical Engineers, 6 December 1934. For an abstract see "Combustion," December, 1934.

† Chemical Engineer, U. S. Naval Engineering Experiment Station, Annapolis, Md.

‡ A New Boiler-Water Treatment for the United States Navy, by T. A. Solberg and Robert C. Adams, Jr., "Combustion," December, 1933.

the classes of the reporting ships and the number of reports received. These ships were chosen to include some of the ones on which there previously had been the greatest difficulty, as well as to include representatives of the various classes which make up the Fleet.

There was one difficulty in the interpretation of the reports which resulted from a confusion in the minds of some of the reporting officers. At the end of the first year of the investigation, in an effort to give an immediate improvement, instead of necessitating a delay of an additional year for the first benefit of the investigation,

TABLE I.

TABULATION OF REPORTING VESSELS.

Class of Ship	*Number Reporting*	*Tons*	*Number of Reports*
Battleship	13	394,500	33
Oiler*	4	60,900	11
Auxiliary	2	48,000	6
Aircraft Carrier	1	33,000	5
Tender	2	22,600	3
Store Ship	2	21,100	6
Destroyer†	15	17,250	40
Repair Ship	1	10,620	2
Hospital Ship	1	9,800	1
Cruiser	1	9,100	6
Minesweeper	9	7,560	20
Tug	4	4,120	9
Total	55	638,550	142

the proportion of tri-sodium phosphate in the boiler compound was increased from ten to almost fifty per cent, and the soda ash and tannin percentages were decreased. Because of the necessity for expending the available stocks of the old compound, some ships did not obtain the modified compound until the fall of 1933. In August, 1933, the determination of the final formula, Navy Boiler Compound 1933, was announced. Several of the later ships to re-

* One oiler made only one partial report.
† One destroyer reported only on an occurrence of severe priming.

ceive the modified compound believed it to be Navy Boiler Compound 1933 and reported accordingly. This misapprehension which was very difficult to detect in reports, persisted in some cases as late as September, 1934, by which time virtually all ships had been supplied with Navy Boiler Compound 1933.

The officers of the selected ships were asked to reply to six specific questions on the effect of the new method and also to give such additional information as seemed to them to be pertinent. The six questions were:

(*a*) What are the conditions as to scale formation or the removal of scale already in the boilers?

(*b*) Is there difficulty with formation of sludges and/or tendency to cloud gauge glasses?

(*c*) Is there any tendency toward priming?

(*d*) What is the effect on corrosion?

(*e*) What method is used in treating feed water and in making chemical analyses of boiler water?

(*f*) Are any difficulties experienced in the control of alkalinity?

The results in service, reported from the use of the new compound and equipment, will be summarized under the headings of the six questions which were asked the commanding officers of all the ships. The officers of Naval vessels are burdened with such a quantity of routine paper work that voluminous replies to an extra questionnaire could not be expected. Some of the officers answered each question in the minimum number of words, but fortunately there were several who were able to prepare very pertinent and informative reports.

SCALE FORMATION AND SCALE REMOVAL.

With a single exception, all ships which reported to the Station stated that the use of the compound either has decreased the rate of scale formation or has stopped scale formation altogether. Since some of these reports unwittingly dealt with the modified compound, it is obvious that the modification was a step in the right direction. The ships which are known to be using Navy Boiler Compound 1933 also record most satisfactory removal of the scale which was deposited previously. These records vary from "a few

pieces of scale found in the cross box," and " pieces of scale as large as three inches across loose in the drum " to " a bucket-full of dislodged scale removed from mud drums after 700 hours steaming."

The single ship which reported an increase in scale formation is a heavy cruiser which for two or three years prior to August, 1933, had used a proprietary material. As soon as the use of proprietary materials was discontinued and all ships were ordered to use the standard equipment and materials, this ship reported the formation of scale toward the burner end in all boilers. Many tube sections, samples of boiler water, and reports have been sent from the vessel to the Station. Careful examination of all the information and material provided indicates that the scale formation occurred prior to the change in treating methods, but that it was not discovered because of the uniform scale thickness and because upon being brushed the scale assumed a deceiving metallic appearance. Upon the establishment of the positive chemical attack provided by Navy Boiler Compound 1933, disintegration of the scale commenced, resulting in unevenness in the scale surface, which permitted detection of its presence. When this ship made its report at the end of September it was apparent that marked progress had been made in scale removal so that scale formation could not be in progress.

FORMATION OF SLUDGES AND/OR TENDENCY TO CLOUD GAUGE GLASSES.

There was considerable variety in the replies to this question. The question was poorly worded and rather confusing since some officers thought it referred to objectionable sludges in the boiler water and others reported on sludges found in the boilers when they were opened. The amount of sludge in the boilers was slightly greater than had been found after the use of the earlier compounds. One destroyer pointed out that while there was a large amount of sludge in the bottom blow, there was none in the feed lines or feed heaters. Another remarked in a recent report that there had been present a large amount of sludge when the boilers first were inspected after the use of Navy Boiler Compound 1933 had been started. At subsequent inspections this amount of sludge had been decreasing, which indicated gradual removal of the scale.

Over three-fifths of the ships reported no trouble from clouding of gauge glasses. Twelve ships characterized the clouding experienced as " slight " or " unobjectionable." A few ships reported the occurrence of clouding immediately after boilers had been cleaned and refilled or clouding which required an occasional bottom blow to clear up, and two ships stated that clouding became so severe that boilers finally had to be dumped. The most significant reports came from two ships which have used Navy Boiler Compound 1933 for almost a year. These ships stated that rather severe clouding in the early months of use of the new compound appeared to be decreasing steadily as old scale deposits were removed.

TENDENCY TOWARD PRIMING.

The ships were virtually unanimous in reporting the absence of any tendency toward priming. One destroyer reported priming when the new compound was first used, but that several bottom blows to remove suspended solids and reduce the alkalinity stopped the trouble. Two minesweepers, which operate in the same squadron, simultaneously sent in identical reports of the occurrence of excessive priming unless the alkalinity were reduced to 0.38 per cent normal. Subsequently one of these ships was found to have a loose dry pipe, which was the cause of the carry-over diagnosed as priming; no explanation for the priming on the other ship has been offered. Another destroyer experienced sufficiently severe priming to force it from formation as a result of needlessly great dosage of compound and consequent too rapid removal of scale. This ship did not have a new boiler-water testing-cabinet and so did not have complete information as to boiler-water composition.

In contrast to these four instances of priming, three of which cannot be attributed with certainty to the use of Navy Boiler Compound 1933, several quite favorable reports have been received in random comments from other vessels. A minesweeper and a cruiser remarked that there had been no trace of priming on full-power trials; two destroyers that none had been experienced at any speeds, even at thirty-four knots; and a battleship reported no priming even with the water level above three-fourths of a glass. This indicates that the new compound certainly does not induce priming, and probably acts to reduce it.

EFFECT ON CORROSION.

In the matter of corrosion prevention, the new methods of treatment have received an approval much more overwhelming than was expected. Thirty-two ships reported a decrease in corrosion of boiler interiors over that previously experienced. Some of them qualified their comment by saying "slightly less." All of the destroyers were in this first group. Nineteen ships reported slight corrosion without making a comparison, or reported no change in corrosion. Three ships failed to report on corrosion, and only one stated that corrosive attack had increased since the change to revised methods and means of control.

The ship which made the lone adverse report is a very old, demilitarized vessel which probably suffers in efficiency from a severe reduction in complement. A comprehensive report on the nature and location of the corrosion showed it to be the result of dissolved oxygen in the feed water selectively attacking the metal first to be freed of the old scale-coating. Corrective measures for preventing access of air to the boilers were suggested by the Station.

METHODS USED IN TREATING AND IN ANALYZING BOILER WATER.

The replies to this question show a decided effort on the part of ships' personnel to avail themselves of the new equipment and with it to maintain closer limits of control on the boiler-water composition. Instead of being satisfied with the alkalinity range of 0.4 to 0.7 per cent of normal allowed by the revised regulations, the reports from thirty-seven, or about two-thirds, of the reporting ships indicate an effort of the personnel to maintain closer control, preferably at the higher limit of alkalinity. Thirteen ships submitted routine, non-illuminative reports such as "treatment according to Manual" but without details showing how close the control was maintained. Since full details are provided in the Manual of Engineering Instructions, some of these thirteen ships probably are controlling boiler-water conditions with greater precision than was indicated by the brief reports. The remaining five reporting ships either offered cryptic comments which could not be interpreted or did not answer the question.

The need of individual treatment for the water in each boiler seems to be becoming more generally recognized throughout the Service, as evidenced by the reports of the various ways in which compound is added to the boilers. The feed arrangement on Naval vessels is such that any feed pump can be used to feed any or all boilers and the piping is fitted in duplicate: main feed-system and auxiliary or emergency feed system. The Station suggested that doses to individual boilers be administered by means of the auxiliary feed-pump instead of by the former drip tank over the hot well, which doses each boiler according to its present steaming rate instead of according to the condition of its water. Many ships use the system suggested although it requires much flushing of lines and considerable extra effort. Some ships which have several boilers and do not use all at all times rotate the idle boilers and add doses to them through the air cocks. The enterprising engineer-officer of one battleship has made up a portable hand pump with which a solution of compound may be pumped from a pail directly into a drum of the needy boiler. Other ships report similar equipment as being designed or under construction.

Few ships have reported on the use of the new boiler-water testing-cabinet because none of the cabinets were received by ships before the arrival of the Fleet in the Atlantic early in the summer of 1934, and some ships which remained in the Pacific had not received their cabinets by October first. The comments from the few ships which have remarked on the new cabinet have been to the effect that it is highly satisfactory.

DIFFICULTIES IN THE CONTROL OF ALKALINITY.

Thirty-six ships reported that they had encountered no difficulty with the control of alkalinity, and ten more presumably had no trouble since they failed to report it. The difficulties reported by the other nine ships were most illuminating. In general they indicated that the Naval boilers contained scale which Navy Boiler Compound 1933 attacked rapidly. A repair ship, a cruiser, and a minesweeper remarked on the rapid decline in alkalinity immediately after a boiler had been cleaned and refilled. This was interpreted by the Station to mean that the compound, which provides the alkalinity, was expended in decomposing scale already in the boiler.

In furtherance of this belief a battleship reported that two or three days were required in which to build up or adjust the alkalinity of a refilled boiler; a hospital ship stated that much closer attention and almost daily doses were required to maintain boiler-water alkalinity; and a destroyer said that compound had to be added to the boilers during a long rotating-reserve period when they were not steamed.

Some of the larger ships described their troubles in maintaining close control of alkalinity when boilers are steamed intermittently. The engineer officer who made up a hand pump for feeding compound did not have this difficulty. It may be that some such arrangement must be provided, either by general fiat or by the exercise of individual ingenuity and initiative.

Three ships commented upon their difficulty in attaining zero hardness of the boiler water. It was to be expected in cases which required the removal of heavy scale that the phosphate of the compound would be expended first, leaving the alkalinity of the boiler water within the proper range but still without complete protection against scale formation. In such cases the recommended remedy, wasteful though it may seem, is to blow down, in order to remove the excess alkalinity, and then to add more compound, repeating the cycle until zero hardness is attained. The same repair ship which noted the initial decline in alkalinity afterward had difficulty in maintaining zero hardness. Her engineer officer followed the recommended procedure throughout most of a three-thousand-mile voyage, although it required an expenditure of almost seventy-five per cent more make-up feed-water. He expressed his complete satisfaction with the manner in which the boilers were becoming clean, and stated that the extra requirement of make-up feed-water had decreased within a month to only twenty-five per cent more than normal. This extra feed-water he characterized as "well-spent, in view of the generally healthly condition of the boilers."

GENERAL COMMENT.

In addition to the replies to the six listed questions, about one in three of the engineer officers offered additional information and comment. Only two were at all unfavorable. One battleship reported a tube failure in each of two boilers during the month of

April, when the new compound first was put in use, but that ship has submitted no subsequent unfavorable reports. A hospital ship complained at being unable to obtain Navy Boiler Compound 1933, except by borrowing from a battleship, as late as July, 1934.

Although all other comments were favorable, those from three ships were confined to a simple statement of the fact of the superiority of the new method over the old. One officer even estimated the improvement as "roughly thirty per cent"! In addition to these three, two other engineers expressed the opinion that the present maximum steaming period of seven hundred hours between boiler cleanings easily could be extended. One, from a destroyer, proposed an immediate extension to one thousand hours.

Five ships remarked upon the amount of old scale removed. One destroyer engineer expressed the belief that a second, seven-hundred-hour steaming period would complete the removal of all old scale.

The greater ease in periodic cleaning of boilers treated by the new methods was commented upon in reports from six ships. Some noticed that it became easier to push rotating brushes through the tubes, and others that a brush could be used to clean many more tubes before it required replacement, while two minesweepers and a battleship remarked that the total time for cleaning boilers had been greatly reduced. One battleship made the comment that virtually all deposit in tubes and drums could be washed out with a hose; another stated that wiping with a soft cloth of the interior surfaces of hand-hole plates and headers disclosed the wire-brush marks of a previous cleaning.

CONCLUSION.

The most noticeable void in the reports from the ships, which have been summarized above, is the lack of information and details of the chemical analysis of boiler water and the control of its composition. This was caused by the unfortunate delay in delivery of the newly designed cabinet, and by the generally damaged condition in which the glassware of the cabinets was found upon delivery.

The former testing-cabinet contained a bottle each of nitric acid and of silver-nitrate solution; an open-top, straight-tube burette for

each solution; a porcelain casserole to hold it during the titration; and a few stirring-rods. Directions for making up the reagents from concentrated acid or from crystalline silver-nitrate were given in the Manual. On large ships the solutions could be made up in the Sick Bay, but small ships usually required the services of a Navy Yard Chemist. The reagents used were of such high concentration that serious errors in analyses resulted from slight inaccuracies in titration. Small ships, particularly destroyers, had much difficulty in making analyses during rough weather because of the virtual impossibility of filling with reagent a burette which swayed through an arc of twenty to fifty degrees.

The boiler-water testing-cabinet was re-designed to make each analyst independent of outside assistance in making up solutions, to increase the attainable accuracy of analyses, to make possible analyses under all weather conditions, and to provide more complete information from the results of analysis. An illustration of the new cabinet is shown in Figure 1. During this re-design it was necessary to consider the training and instruction which probably would be common to the boiler-water analysts who would use the cabinets.

The analysts, or Water Kings as they are designated in the Navy, usually have had at least four years of Naval service and their previous training sometimes includes a high-school education. Their instruction in water analysis is derived either from personal direction by an officer or from individual study of the Manual of Engineering Instructions.. The officers themselves, except in rare cases, have had no chemical training beyond the necessarily rudimentary amount which is crowded into the course at the Naval Academy, so that their primary source of information is the same Manual. Since, as his title indicates, the Water King usually is a rather self-sufficient individual, the best-advised plan was to increase that self-sufficiency and individual responsibility by designing simple testing-equipment and providing written directions which could be easily used and completely understood by a seaman of average intelligence.

The reagent bottles in the new cabinet have a one-liter, volumetric marking, and reagent chemicals now are supplied as standard stock solutions, twenty or ten times reagent strength. An exhausted

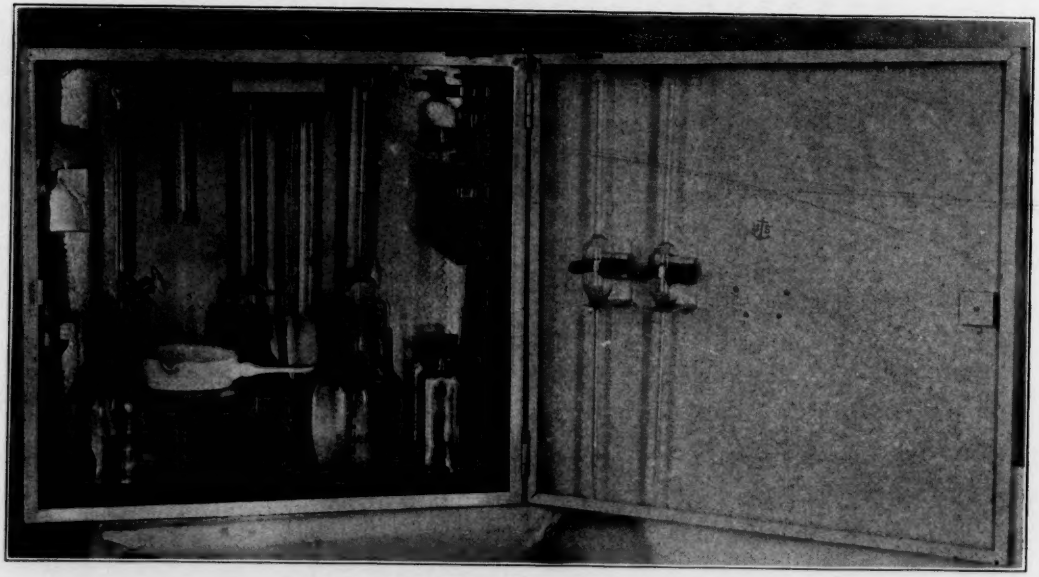


FIGURE 1.—NAVY BOILER-WATER TESTING CABINET.

supply of reagent is replenished by transferring the necessary fifty or hundred milliliters of stock solution to the reagent bottle with the pipette or graduated cylinder provided in the cabinet, and then diluting to the mark.

The concentration of the reagents and the size of the burettes were reduced simultaneously to provide more accurate analytical determinations. The reduced concentration of the reagents was so chosen that the burette reading, after either multiplication by two or subtraction of a previously determined constant, could be entered directly in the boiler-record sheet as per cent normal alkalinity or as grains per gallon.

A bottom-filling, automatic-zero burette is fitted through the rubber stopper of each reagent bottle. The burette is filled by air pressure on top of the solution supplied by a small, rubber aspirator-bulb. When the burette is full, the pressure is released, and the reagent level drains to the zero mark. The analyst has only one burette reading to make, at the end of the titration, and in place of the former three chances of error, two readings and a subtraction, he now has only one.

Soap solution, for the determination of hardness, was introduced as a third reagent. Stock soap-solution twenty times reagent strength is more concentrated than can be made up, so that the stock soap-solution supplied to the Fleet is only ten times reagent strength. The dilution is made with alcohol which already is carried in ships' stores. The lather factor of each bottle of reagent soap-solution must be determined immediately after the dilution is made. The amount of this lather factor is subtracted from the burette reading after each hardness determination and the balance is the hardness of the sample in grains per gallon of calcium carbonate.

Hardness was chosen as the supplementary determination from alternative determinations for several reasons. Its purpose was, of course, to measure the actual scale-combative chemical present in the boiler water. This it does not do directly, since carbonate and phosphate are the actual chemicals, but a water which has zero hardness may be inferred to have sufficient carbonate and phosphate present. The phenolpthalein-methyl-orange titration for carbonate is most unreliable especially so when made by an untrained analyst.

More accurate carbonate determinations either require extensive equipment or are so changed from simple titrations as to render the results questionable in the hands of a Water King. All methods for the determination of phosphate are so different from simple titrations that they are open to the same objection. Finally, the determination of either carbonate or phosphate is only half an answer, since in the absence of either the other still is effective.

The non-corrosive condition of boiler water treated with Navy Boiler Compound 1933, as almost unanimously reported from the Fleet, was somewhat surprising. At the time that the new treatment was recommended for the Service, it was expected that reports similar to the single adverse one might be the rule. It was known that the ships of the Navy did not have de-aeration equipment at all comparable to those now provided for shore power plants of equivalent size. There had been no appreciable difficulty from corrosion because of the protection afforded by scale coatings which existed in virtually every boiler. It was feared that the scale removal effected by Navy Boiler Compound 1933 would expose the boiler metal to corrosive attack. Several precautions in the care of feed water were described in the Manual of Engineering Instructions to guard against severe attack of this nature. In spite of these mechanical precautions there was the possibility that the use of some chemical reducing-agent might have to be included in the boiler-water treatment. Fortunately the use of oxygen-reducing chemical, with its attendant control and increased complexity of boiler-water treatment, now appears to be unnecessary.

In conclusion, all of the evidence which the reports from the Fleet have provided indicates that the new method of treatment has been fully as satisfactory and successful as had been hoped. There have been minor difficulties, such as may be expected in any organization of similar size, but the remedy for each has been supplied readily. It would appear that the Navy will not require another major modification of its methods of boiler-water treatment for many years.

PROBLEMS OF ANTI-ROLLING STABILIZATION OF SHIPS BY THE ACTIVATED TANK METHOD.

By Dr. Nicholas Minorsky.*

1. Introduction.

Attempts to stabilize ships against rolling made during the last half century are well known (1)†; however, it appears desirable to mention briefly a few principal methods of ship stabilization which have been actually tried in practice.

(*a*) *Water ballast stabilization, originally invented by Sir Ph. Watts* (2) *in England and, later on, improved by Dr. Frahm* (12) *in Germany:* This method is generally known under the name of "Frahm's Tanks" or "Passive Tank System." A number of ships (mainly German) are still equipped with Frahm's tanks. The main advantage of this method lies in its simplicity and low cost; its disadvantages are a comparatively low amount of roll-quenching and an erratic behavior when the sea is not of regular trochoidal pattern.

(*b*) *Gyroscopic method of stabilization, originally invented by Dr. O. Schlick in Germany and later on improved in the United States by the late Dr. E. A. Sperry* (3): For a number of years this method was applied by the Sperry Gyroscope Company for stabilization of numerous ships, of which the largest one equipped with stabilizers of this kind is the Italian liner *Conte di Savoia* of the Lloyd Sabaudo Line. The main advantage of this method consists in its high roll-quenching efficiency when the stabilizer works within its roll-quenching range; the fundamental drawback is its excessive cost, especially for larger vessels, and also a comparatively large net weight of installed equipment.

(*c*) *Outboard methods of stabilization based on the principle of a suitable operation of movable keels or vanes by a control mechanism:* One apparatus of this kind was developed by Motora in

* Mechanical Engineering Department, Massachusetts Institute of Technology.

† Numbers refer to Bibliography Notes at end of article.

Japan (4) and the other by Kefeli in France. The principal advantage of this method is in its low weight. There are, however, very strong objections to the use of such outboard controlled vanes on the part of naval architects and constructors, partially from the standpoint of complications in the design of the hull and partially on account of an increased resistance to propulsion which they offer.

(*d*) *Solid moving weight method of stabilization,* originally suggested by Sir J. I. Thornycroft in England (5) and tried out on a small vessel: On this basis certain later improvements were made by Norden in the United States (6) and by the Siemens & Halske Co., (6′) in Germany. With a suitable control the method is capable of producing a comparatively high roll-quenching efficiency. The main drawback of the method consists in the necessity to arrange in the ship a transverse passage of sufficient cross section to accommodate the tracks for the weight. This generally entails extensive changes in the ship's design. The safety requirements in case the weight becomes loose on its track is another objection to this method of stabilization.

(*e*) *Controlled water ballast method of stabilization, generally designated as activated or active tank method:* As the name implies, this activated system differs from the earlier passive tank system in that the flow of ballast between the tanks, instead of being governed by ship's rolling motion, is actuated by a local source of power (*e.g.*—air pressure, impeller pump, etc.) controlled by instruments responsive to rolling. This method was known for many years, but on account of the lack of an adequate control no progress was made until recently when it began to attract the general attention of naval architects and constructors by its potential possibilities. It is apparent that if the control problem is solved correctly, this method combines the advantages of the passive tank system as well as of the gyroscopic method without having their respective disadvantages. In many cases the weight of the ballast may be incorporated in the useful weight which the ship has to carry anyhow (*e.g.,* oil, fuel or fresh water supply), and the only weight charged to the stabilizer proper, in such a case, is the weight of the power plant producing the actuation of ballast, and this, generally, amounts only to a fraction of 1 per cent of the ship's displacement. In case outboard water is used as ballast, it can easily be

pumped out when there is no need for stabilization. Finally, on naval vessels the tanks can be incorporated into anti-torpedo protective blister system so that no useful space is lost. One can thus recognize a series of very important features rendering the application of this activated tank method particularly attractive, both for naval and merchant vessels.

It is proposed in the following to investigate the fundamental principles of performance of a theoretically perfect activated tank stabilizer and in this way to gain an insight into the problem of a control capable of producing such stabilization. It will be apparent from this study that the choice of controlling system under these requirements is narrowed down to such an extent that it is possible to specify definite characteristics which such control system must possess in order to insure that the roll-quenching efficiency is at all times at its maximum consistent with the amount of the ballast employed as well as with displacement, metacentric height and period of the ship. It is of interest to note that the theoretical conclusions reached by the following analysis have found a complete experimental confirmation (7) during the experiments which have been carried on recently with a model of 2000 pounds; this model was arranged to represent, on a scale reduced by the law of similitude, the performance of a normal ship in a regular as well as an irregular seaway. It is hoped that the following investigation will be of interest to those concerned with these problems, not so much from the standpoint of certain particular constructional features, but rather from the more general viewpoint of the principles underlying the behavior of an active tank stabilizer whatever its design may be.

The engineering problems connected with the design of an active tank system are not treated here, and will be published at a later date.

2. PROBLEM OF SHIP STABILIZATION.

Within the limits of approximations assumed by Wm. Froude in his theory of resisted rolling among waves, the differential equation of rolling among waves is (8):

$$I\frac{d^2\theta}{dt^2} + k_1\frac{d\theta}{dt} + k_2\left(\frac{d\theta}{dt}\right)^2 + Wh\theta = \varepsilon Wh\Theta \operatorname{Sin}\omega t \quad (1)$$

where θ is the angle of rolling

W is displacement of the ship

h is metacentric height

I is moment of inertia of the ship about longitudinal axis through the center of gravity

Θ is the maximum angle of the effective wave slope; k_1, k_2 coefficients of natural resistance to rolling (*e.g.*, skin friction, bilge keels, etc.), assumed by Froude to be proportional to the first and to the second power of angular velocity of rolling (9).

ε is empirical coefficient compensating for approximations of Froude's theory; in many text books of naval architecture it is assumed that $\varepsilon = 1$. This will be assumed also in the following.

ω is the apparent period of the waves.

Approximations of Froude's theory are well known and need not be discussed here. Equation (1) presents a considerable difficulty from the mathematical standpoint on account of the term with k_2.

It is possible to obtain an approximate solution of equation (1) either by the method of successive approximations (10), or graphically, as suggested by Froude (11).

For investigation of a quantitative behavior of the stabilizer it is necessary to solve a system of two differential equations (12), one expressing the action of stabilizer on the ship, and the other the reaction of the ship on the stabilizer.

In the passive tank system there is no external power provided to actuate the stabilizer and this actuation is produced solely by the reaction of ship on the stabilizer. In other words, the residual rolling of the ship produces both the potential and the kinetic reactions necessary for displacement of the water ballast athwartships.

In view of the fact that the theory of the passive tank stabilizer has been studied very thoroughly (13), it will not be reproduced here.

A rigorous study of the active type stabilizer must develop along the same line of argument.

It will be shown later that, owing to the active principle, a gravity stabilizer has generally a much greater accuracy of stabilization

and the potential heads due to residual rolling are considerably reduced on that account.

There remains, however, a kinetic reaction from the stabilizer on the ship. In the following study, as an example, the actuation of the ballast is produced by means of an impeller coaxial with the channel connecting tanks. In this particular case the kinetic reaction is the thrust from the impeller applied to the ship athwartships. The moment of this reaction depends upon the position of the thrust bearing with respect to the center of gravity of the ship. When the line of this reaction goes through the center of gravity its moment is zero and the only moment applied to the ship from the stabilizer is that due to the moment of the transferred ballast.

Assuming, furthermore, as a limiting case, that the stabilization is accurate, we can neglect the passive component of the stabilizing performance and the whole problem mathematically reduces itself, under those conditions, to the modification of the original Froude's equation by adding to the left side a new term S, characterizing the stabilizing moment. The second equation, under such approximations, is not necessary.

The differential equation of a stabilized ship is:

$$I\frac{d^2\theta}{dt^2} + k_1\frac{d\theta}{dt} + k_2\left(\frac{d\theta}{dt}\right)^2 + S + Wh\theta = Wh\Theta \sin\omega t \quad (2)$$

It must be noted that equations (1) and (2) both have harmonic second members as was assumed by Froude. If the sea is erratic the clearly harmonic second member cannot represent such condition and must be replaced by a certain unknown function of time $f(t)$ which, generally speaking, may not be periodic and represents action on the ship of isolated impulses due to subsequent waves.

The question arises as to whether or not the stabilizing moment S should be included in the left member of equation as given by equation (2), thus representing the still water rolling of the ship, or must be referred to its right member characterizing the disturbance. In the first case the action of the stabilizer may be designated as *damping,* in the second as *compensating.* Attempts were made (14) to classify stabilizers according to this difference of their action. It will be shown later that this difference is not essential and that the stabilizer based on compensating action possesses at the

same time the damping action and vice versa. This will be more apparent from a consideration of performance in the erratic seaway (section 4).

Provisionally, we will assume that the stabilizer is based on damping action (equation 2) and, since all terms on the left are continuous functions of time, clearly S must be also a continuous function.

An ideal stabilizer based on damping action must thus have a feature of continuity; this point is of a considerable importance for the following reasons and will be referred for the sake of abbreviation as the "principle of continuity" of the control. A stabilizer having a continuous action will always be more efficient than the corresponding stabilizer not having this feature because in the former the stabilizing damping action will follow the disturbing moment of the waves by continuity while in the latter no such balance between the cause (waves) and the effect (stabilizing moment) is possible in instantaneous values but only as an average which is not sufficient, especially for a confused seaway. We are now in a position to establish the functional relation between the stabilizing moment S and the rolling motion which it is purposed to quench.

In fact, rolling, especially over large angles, is an essentially cumulative process, each wave impulse contributing an *increment* to that which has already been built up by preceding waves. Hence the stabilizer's action must on the contrary manifest itself in production of suitably timed *decrements.*

The natural decrements, however, exist in any ship and are measured by coefficients k_1, k_2 of equations (1) or (2). The best way of defining the problem of ship stabilization, perhaps, is to define it as a method of artificial increase of the value of coefficients k_1 and k_2 in Froude's equation (1) for a given ship (15).

From the foregoing it is immaterial as to whether k_1, k_2 or both are increased by action of the stabilizer.

From the practical standpoint, it is simpler to design a stabilizer the moment of which varies in proportion to the angular velocity $\frac{d\theta}{dt}$ of rolling than one in which this moment varies in proportion to the square of this velocity.

In addition to this the mathematical analysis of the problem and hence the predetermination of characteristics are considerably sim-

plified because k_2 can, in the first approximation, be neglected in comparison with $K = k_1 + K_s$ where K_s is the equivalent coefficient characterizing the action of the stabilizer, which may be made considerably larger than k_1 due to the natural damping (skin friction, bilge keels) proportional to the first power of angular velocity. There is another reason why k_2 can be neglected in the following due to the fact that, if the steady state of stabilization is already established, the amplitudes of oscillation are generally small, as well as the angular velocity of rolling motion, hence $\left(\frac{d\theta}{dt}\right)^2$ is of the second order of small quantities and can be neglected for that reason.

The mathematical analysis of the motion of a stabilized ship under these conditions becomes considerably simpler than that of the non-stabilized ship given by equation (1).

The differential equation for a stabilized ship is (16):

$$I\frac{d^2\theta}{dt^2} + K\frac{d\theta}{dt} + Wh\theta = Wh\Theta \operatorname{Sin}\omega t \quad . \quad . \quad . \quad (3)$$

Equation (4) is obtained by dividing equation (3) by I and by putting

$\frac{K}{I} = a$; $\frac{Wh}{I} = b^2$. Designating for abbreviation $\frac{d^2\theta}{dt^2} = \ddot{\theta}$ etc. we have finally:

$$\ddot{\theta} + a\dot{\theta} + b^2\theta = b^2\Theta \operatorname{Sin}\omega t \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The general solution of equation (4) is

$$\theta = Ae^{-r_1 t} + Be^{-r_2 t} + b^2\Theta\sqrt{\varphi}\, e^{i\omega(t-\tau)} \quad . \quad . \quad . \quad (5)$$

where A and B are the constants of integration to be determined by initial conditions, r_1 and r_2 are the roots of the, so called, characteristic equation

$$r^2 + ar + b^2 = 0 \text{ and } \varphi(\omega) = \frac{1}{(b^2 - \omega^2)^2 + a^2\omega^2} \text{ which} \quad . \quad (6)$$

is sometimes designated as the *resonance function.* Finally $\operatorname{Tan}\Psi = \operatorname{Tan}\omega\tau =$

$$= \frac{a\omega}{b^2 - \omega^2} \quad . \quad . \quad . \quad . \quad (7)$$

The steady state of rolling among harmonic waves is obtained by observing that the first two terms in equation (5) gradually die out and there remains only the last term, called sometimes, *forced oscillation* θ_1 which can be written also as

$$\theta_1 = \frac{b^2 \Theta}{\sqrt{(b^2 - \omega^2)^2 + a^2 \omega^2}} \quad . \quad . \quad . \quad (8)$$

Equation (8) gives the amplitude of the residual quenched oscillation; its phase relative to the wave slope Θ is given by equation (7).

At synchronism between the periods of the ship $T = \frac{2\pi}{b}$ and the waves $T_1 = \frac{2\pi}{\omega}$, $\omega = b$ and

$\theta_{1s} = \frac{b^2\Theta}{a\omega} = \frac{bI\Theta}{K}$ (9) and the phase angle $\psi = \text{Tan}^{-1} \frac{a\omega}{b^2 - \omega^2} = \frac{\pi}{2}$

the residual quenched oscillation lags behind the wave slope by one quarter period.

Substituting in (8) $b = \frac{2\pi}{T}$ and I from equation $T = 2\pi \sqrt{\frac{I}{Wh}}$ (17)

giving the free period of the ship in terms of its parameters we have

$$\theta_{1s} = \frac{T}{2\pi} \frac{Wh\,\Theta}{K} = \frac{Wh\,\Theta}{K\,b} \quad . \quad . \quad (11)$$

Where θ_{1s} is the amplitude of the residual quenched oscillation at synchronism. Using the well-known relation between the coefficient of damping k_1 and the decrement a (15) and defining the decrement A of a stabilized ship in the same manner in which the decrement *a* due to bilge keels is defined we have:

$$A = \frac{K \pi^2}{WhT} \quad . \quad . \quad . \quad . \quad (12)$$

in which $K = k_1 + K_s$ already contains the stabilizer's damping by the term K_s.

Substituting K from equation (12) into equation (11) we have

$$\frac{\pi}{2} \Theta = A\, \theta_{1s} \quad . \quad . \quad . \quad (13)$$

The left member of equation (13) represents the synchronous increment per roll of the ship due to the wave slope Θ; the left member of the same equation represents the decrement produced by the stabilizer for the synchronous amplitude θ_{1_s} of the forced rolling. For sake of abbreviation we will designate A as *stabilizing decrement* of the stabilizer.

The process of anti-rolling stabilization in this manner can be expressed in the case of synchronism as equilibrium between the synchronous increment $\frac{\pi}{2}\Theta$ due to the wave slope and the *stabilizing decrement* A characterizing the roll quenching capacity of the stabilizer.

For a given wave slope Θ, the stabilized amplitude θ_{1_s} of residual rolling will be the smaller the larger the stabilizing decrement A. The degree of accuracy of stabilization will be thus entirely determined by A.

If the stabilizer is stopped, the synchronous rolling will develop and the equilibrium amplitude θ_1 will be reached when the synchronous increment $\frac{\pi}{2}\Theta$ becomes equal to the decrement due to bilge keels and skin friction, which may be designated as " natural " damping to distinguish it from the *stabilizing damping* A with which, thus far, we have been concerned. We have thus (17):

$$\frac{\pi}{2}\Theta = a\theta_1 + b\theta_1^2 \quad . \quad . \quad . \quad (14)$$

The amplitudes reached in case of non-stabilized synchronous rolling are generally considerable and coefficient b cannot be neglected as we have done in case of stabilization.

Solving equation (14) for θ_1 one can compare directly the stabilized θ_{1_s} and non-stabilized θ_1 amplitudes and to predetermine thus the amount of roll quenching provided coefficients a, b and A are known.

The first two coefficients can easily be determined from the curve of declining angles (18).

The determination of the coefficient A relates to the design of the stabilizer.

We propose now to establish an explicit relation between the coefficient K_s and the parameters of the gravity stabilizers. To make this study more general we will not discriminate between the solid moving weight stabilizers and those utilizing water or oil transfer between the tanks.

Designating by θ the instantaneous angle of roll we have

$$\theta = \theta_1 \text{ Cos}\omega t \text{ and } \dot{\theta} = -\theta_1 \omega \text{Sin}\omega t \text{ and}$$

In view of the fact that the stabilizing moment S should be of the form $S = K_s \frac{d\theta}{dt}$ we have:

$$S = -K_s \theta_1 \omega \text{ Sin}\omega t \qquad (15)$$

For harmonic motion $S = S_o \text{ Sin}\omega t$ where S_o is the maximum static moment of the stabilizer. Hence as far as the amplitude is concerned:

$$K_s = \frac{S_o}{\theta_1 \omega} \qquad (16)$$

Substituting in equation (12) k_1 and K_s from equations (11) and (11′) and rearranging we have:

$$\theta_1 a = \frac{\pi}{2} (\Theta - \Theta_s) \qquad (17)$$

where $\Theta_s = \frac{S}{Wh}$ and S is the maximum value of stabilizing moment. Equation (17) is equivalent to equation (13). It expressed the same fact as equation (13) namely, that the equilibrium amplitude θ_1 of a stabilized ship will be reached when the *natural decrement* $\theta_1 a$ associated with this amplitude θ_1 becomes equal to the *stabilizing increment* $\frac{\pi}{2} (\Theta - \Theta_s)$ at synchronism.

By *stabilizing increment* is meant not the increment due to the actual effective wave slope Θ but to a resultant wave slope $\Theta - \Theta_s$ obtained by subtracting from Θ the slope $\Theta_s = \frac{S}{Wh}$ equal to the static angle of list Θ_s which the stabilizer can produce in still water when a given stabilizing moment S is developed.

It will be shown in the following that it is possible, generally speaking, to change the amplitude S of stabilizing moment continuously, if the controlling devices are based on continuous action.

Thus, if controlled tanks are used, more or less water can be transferred between the tanks according to the intensity of the control. It is evident, however, that there is a maximum S_0 above which the stabilizing moment can not go; in case of the tanks this will happen for example, when one tank is full and the other empty; in case of the solid weight this will occur when the weight has reached its extreme position.

In order to take into account the possibility for such adjustment of the stabilizing moment assume that it is governed by equation:

$$S = S_0 (1 - e^{-\lambda}) \quad . \quad . \quad . \quad (18)$$

where λ is the parameter determining such variation and e is the base of Neperian logarithms. No particular significance should be attached to the form of equation (18) except that it fulfills the boundary conditions:

for $\lambda = 0$ $S = 0$ there is no stabilization

$\lambda = \infty$ $S = S_0$ there is full control

One can consider λ thus as a certain resistor shunting the electrical control of the stabilizer; in fact when this resistor is zero, the device is shorted and the stabilizer is stopped ($S = 0$); if the resistor is opened ($\lambda = \infty$) the full control is on ($S = S_0$). Between these points the stabilizing action varies by continuity. We have also

$$\Theta_s = \Theta_{so} (1 - e^{-\lambda}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

where Θ_{so} is the maximum list which the stabilizer can produce in still water. Replacing the value of Θ_s from (19) into (17) we have:

$$\theta_1 a = \frac{\pi}{2}\left[\Theta - \Theta_{so} (1 - e^{-\lambda})\right] \quad . \quad . \quad (20)$$

Two important cases must be mentioned:

1. $\Theta \leq \Theta_{so}$, that is the maximum angle of list which the stabilizer can produce in still water is greater (or at least equal to) the maximum effective wave slope encountered.

In such a case it is always possible to find a value for λ (*i.e.*, to adjust the intensity of controlling action) at which $\theta_1 = 0$ theoretically.

Practically this means that the residual rolling oscillation can be reduced indefinitely by increasing the sensitivity of controlling instruments and circuits. The accuracy of roll quenching in this case is governed by the degree of sensitivity of instruments and circuits and not by the stabilizer proper (*e.g.*, amount of water in the tanks, etc.).

Such performance may be designated as *performance of the stabilizer within its roll quenching range.*

2. $\Theta \geq \Theta_{so}$ in this case the maximum list produced by the stabilizer in still water generally is smaller than (or at least equal to) the maximum effective wave slope. In this case even going to the extreme $\lambda = \infty$ $S = S_o$ one has for the increment the value $\frac{\pi}{2} \times (\Theta - \Theta_{so})$. Everything will happen in this case as if, instead of the effective wave slope Θ of the sea being actually applied to the ship, a smaller effective wave slope $(\Theta - \Theta_{so})$ were acting. In this case the ship cannot be stabilized as closely as in the previous case; and increasing the sensitivity of instruments and circuits will not help to reduce the amplitude of residual rolling which will develop exactly in the same manner (with respect to the slope $\Theta - \Theta_{so}$) as if no stabilization were provided.

In this case it is more logical to apply the usual formula (14) for synchronous rolling θ_1 in the form

$$a\theta_1 + b\theta_1^2 = \frac{\pi}{2} (\Theta - \Theta_{so}) \quad . \quad . \quad (21)$$

The action of stabilizer is given by the term Θ_{so} which decreases the action of the wave slope Θ.

The performance of the stabilizer in this case may be designated as that *outside its roll quenching range.* For the sake of abbreviation it is convenient to designate the quantity Θ_{so} as the *roll quenching range of the stabilizer;* it constitutes a fundamental constant for each stabilizer of gravity type and its determination is

immediate once the maximum static moment S_o of the stabilizer as well as displacement W and the metacentric height h of the ship are given.

Finally one particular case of stabilization must be mentioned. When $\Theta < \Theta_{so}$, that is, when the performance is *within* the roll quenching range of the stabilizer, it is possible to make $\lambda > \lambda_1$ where λ_1 is the value for which $\Theta - \Theta_{so} = O$ in equation (20).

Theoretically one cannot readily see what may happen in this case since already for $\lambda = \lambda_1$ the residual motion $\theta_1 = O$.

From the practical standpoint, such a case can be produced experimentally and interesting phenomena observed in this connection are described in the experimental part of this investigation in view of the fact that they do not enter logically into the scope of the theoretical investigation on the basis of the Froude's theory with which we are concerned at present (19).

When the rolling is not synchronous and the stabilizer works within its roll quenching range, the residual forced amplitude θ_1 is changed very little in view of the fact that the term $a^2\omega^2$ can be made large enough in comparison with $(b^2 - \omega^2)^2$.

In case the rolling is non-synchronous and the stabilizer functions outside its range the residual rolling which will develop will be of ordinary non-synchronous type (*e.g.*, with beats between free and forced oscillations) but it will correspond to the equivalent slope $\Theta - \Theta_{so}$ as per equation (21).

Preceding results apply to forced oscillations; in view of the irregularities in the waves encountered, dyssymmetry and other causes, free oscillation (20) reappears from time to time and the first two terms on the right side of equation (5) are not zero after a long time as we have assumed.

The action of the stabilizer in this case manifests itself in the terms $Ae^{-r_1 t}$ and $Be^{-r_2 t}$ where $r_{1,2} = \dfrac{-a \pm \sqrt{a^2 - 4b^2}}{2}$.

If the damping action of the stabilizer is considerable $a^2 - 4b^2 > 0$, both roots r_1 and r_2 are real and negative and the free oscillation is periodically reduced by the stabilizer, as soon as it appears.

If, however, the roll quenching capacity of the stabilizer is not sufficiently great $a^2 - 4b^2 < 0$ and the general solution instead of exponential form can be written more conveniently as:

$Ae^{-\frac{a}{2}t}\cos\left(\frac{\sqrt{4b^2-a^2}}{2}t+\varphi\right)$ where A and φ are again integration constants and $T_1 = 2\pi\sqrt{\frac{4}{4b^2-a^2}}$ is the damped free period of the ship. Coefficient $\frac{a}{2}$ is called the *damping decrement* of free oscillation; the loss of free amplitude per swing in this case is $\exp.\frac{-2\pi a}{\sqrt{4b^2-a^2}}$: free oscillation decreases the faster the greater the value of a, that is, the greater is the damping torque of the stabilizer.

Owing to the increased decrement produced by the stabilizer, free oscillation is wiped out much faster than in case of a non-stabilized ship and the residual rolling of a stabilized ship will consist, practically, of the forced oscillation only.

3. METHODS OF CONTROL.

From the preceding follows that the object of stabilizer is to produce a moment $S = -K_s\frac{d\theta}{dt}$ (22) substantially in proportion to, and in phase with the instantaneous angular velocity or rolling.

The concentration of the water ballast must be a maximum in the tank which is instantaneously rising in space at the fastest rate which generally occurs when the ship passes through the even keel position. When the ship reaches the extreme position and has no angular velocity the amount of water in the tanks must be equal.

Figure 1 shows different phases of ship stabilization in accordance with equation (22) in case of regular rolling, and Figure 2 relates to quenching of an irregular rolling for the sake of example assumed to be of the form shown on figure as θ.

Physical significance of such control can be readily understood from the fact that in all cases the excess of the ballast is on the rising side of the vessel and the work of the wave slope instead of accelerating the ship and building up larger angles of roll is continuously expended in raising an extra weight.

One immediate conclusion can be formed from a mere inspection of the figures, namely, the control at all times must be " down hill," that is, the accelerating power must be applied to the ballast when it is concentrated on the instantaneously rising side of the ship.

This conclusion, of course, can be reached from a consideration of the passive system in which no external power is applied and the flow of the ballast is, to a great extent, due to the potential head caused by the difference in heights of the tanks in space due to residual rolling.

For regular synchronous rolling, the active system merely increases the rate of transfer of the ballast of the equivalent passive system (all other things being equal) but does not change appreciably the phase of the ballast.

For irregular rolling, when the passive system generally drops out of step, and behaves erratically,—the active system forces the phase of the ballast so as to approximate the condition (22) even in this case. This can readily be seen in case of Figure 2.

We thus reach an important conclusion, that, *in all cases* (*regular as well as erratic rolling*) *control of the ballast must be such as to fulfill the condition of equation* (*22*).

Whatever method of control is used for actuating the ballast it is of secondary importance as long as the fundamental condition (22) is fulfilled.

It will be shown now that whereas the condition (22) defines the roll quenching property of the stabilizer in a unique manner, the method of control which satisfies the condition (22) of stabilization is by no means determined by this condition in a single valued manner.

Different methods of control can be produced and still can satisfy the condition (22) of roll quenching. Consider the active tank system. The following line of argument applies as well to the solid moving weight stabilizer by remembering that whereas in the tanks the lever arm L (*i.e.*, distance between the tanks) is fixed and the

weight W is varied, in the solid weight system, the weight is fixed and the lever arm L is changed by the control.

In any control there are two parts to be considered, namely, the primary (or stabilizing) control and the secondary (or follow up) control. The former determines the beginning and the latter the termination of the controlling action.

For the time being we will investigate only the primary control.

The identification of two different expressions for the stabilizing moment gives:

$$K_s \frac{d\theta}{dt} = wL$$

Hence the " transferrable " ballast must be varied so as to satisfy the condition

$$w = a \frac{d\theta}{dt} \quad . \quad . \quad . \quad . \quad (23)$$

where $a = \frac{K_s}{L}$; in other words, the amount W of the ballast concentration must be proportional to, and in phase with the instantaneous angular velocity of the ship. The primary control of the ballast must be derived in this case from an instrument responsive to angular velocity such as a constrained gyroscope suitably positioned to respond to rolling.

If equation (23) is differentiated with respect to time we have

$$\frac{dw}{dt} = a \frac{d^2\theta}{dt^2} \quad . \quad . \quad . \quad (24)$$

Equation (24) gives another possible control of the rate of transfer $\frac{dw}{dt}$ by angular acceleration $\frac{d^2\theta}{dt^2}$ of the ship. It is clear that equation (24) still satisfies the fundamental condition (22) of stabilization after being integrated with respect to time. The primary control in this case must be taken from an accelerometer and applied to control the rate of flow $\left(\frac{dw}{dt}\right)$ of the ballast and not its amount as was done in the first case.

Amount of Water, Port Side
(Angular Velocity θ̇ to Starboard of Vessel)
Angle of Roll, θ° (Starboard)
Velocity of Flow to Port
Acceleration of Flow to Port
Amount of Water Starboard
Velocity of Flow to Starboard
Acceleration of Flow to Starboard
Flow ceases
THEORETICAL PERFORMANCE OF ACTIVE TANK SYSTEM.
Synchronous Rolling
Mar. 10 '32.
N. MINORSKY.

FIGURE 1.

THEORETICAL PERFORMANCE OF
ACTIVE TANK SYSTEM.
Intrepidity Rolling
Mar. 10 '32.
N. MINORSKY
Velocity of Flow to Port $(\frac{dw}{dt} = \dot{w})$
Amount of Water, Port Side
(Angular Velocity $\dot{\theta}$ to Starboard of Vessel)
Angle of Roll, $\theta°$
Acceleration of Flow to Port
Amount of Water, Starboard
Velocity of Flow to Starboard
Acceleration of Flow to Starboard
Flow Cycles
1
2
3
4
5
6
7
8
9
10

FIGURE 2.

By again differentiating equation (24) we have:

$$\frac{d^2w}{dt^2} = a\,\frac{d^3\theta}{dt^3} \quad . \quad . \quad . \quad (25)$$

Which gives another method of possible control, *i.e.*, still satisfying the fundamental equation (22) but in which the device responsive to the rate of acceleration $\left(\frac{d^3\theta}{dt^3}\right)$ controls the acceleration $\left(\frac{d^2w}{dt^2}\right)$ of the flow of ballast.

It is apparent that by extending this method of reasoning one could imagine additional methods of control responsive to still higher time derivatives of residual rolling motion and yet capable of producing stabilizing action in conformity with equation (22).

Theoretically, for a regular sea and negligible time lags these different methods are equivalent insofar as they all satisfy the same equation (22) expressing the condition of anti-rolling stabilization. When the rolling is not regular and consists of a series of impulses due to waves of different lengths, periods, shapes and erratic phases, each of these impulses must be treated on its own merits and the duty of the stabilizer in such a case is to produce the stabilizing moment in accordance with equation (22) rapidly enough to be able to anticipate the disturbing impulse of the wave. In case of such erratic conditions to be mastered the difference between the above mentioned systems of control becomes more definite. Curve of Figure 2 illustrates the above considerations. Curve θ is the actual record of an erratic rolling observed on a ship with a high metacentric height. The curves shown are first (velocity), second (acceleration) and third (rate of acceleration) time derivatives respectively obtained graphically as subsequent slope curves of curve θ, etc.

In vicinity of the point M when the erratic character of rolling appears it can be seen that the second derivative leads the first by approximately 1.5 sec.; the third derivative precedes the second by about .5 sec., and so on. Viewed from this angle the control of velocity of ballast from the accelerometer $\left(\frac{dw}{dt} = a\frac{d^2\theta}{dt^2}\right)$ is capable of following the disturbance at an earlier time than the con-

trol of quantity of ballast from a constrained gyro $\left(w = a\frac{d\theta}{dt}\right)$; but the accelerometer control from this standpoint will be inferior to the acceleration of ballast control in the rate of acceleration control (equ. 25).

Whereas the primary or stabilizing control is expressed in the right member of equations (23), (24) and (25) defining various control methods consistent with the fundamental condition (22) of stabilization, the secondary or follow-up control finds its expression in the left side of these equations. If the power supply were unlimited and the inertia of the transferred ballast could be neglected no special follow-up arrangement would be needed, the ship itself being a kind of secondary follow-up system.

If, however, by practical considerations the capacity of the power plant operating the ballast movements is to be kept as low as possible this power capacity generally becomes of the same order as the power absorbed or regenerated during accelerations or decelerations of the ballast. The follow-up control in such a case is of importance insofar as it permits the use of a rather intensive stabilizing control without its disadvantage which occurs when the ballast acquires too much momentum from the power plant and may get out of step with the disturbance as will be shown later on.

The problem of the secondary or follow-up action in this problem is less apparent than in numerous problems (*e.g.*, gyroscopic compasses, fire control devices, remote indication of levels, direction in space, etc.) where the follow-up system is intended for a purely positional relation between the primary (direction or position indicating) and the secondary (or follow-up) systems.

In the problem of ship stabilization the function of the follow-up secondary control does not concern the geometrical elements of motion but is linked up with the dynamics of the ultimate ship's motion and for this very reason its function is not so explicit. It is true that the ship itself may be considered as a kind of follow-up system and a conclusion could be formed on this basis that no special follow-up system is needed in the control. Experiment indicates that, under certain conditions, good results can be obtained without any special follow-up control. This generally happens when either the disturbing moment is regular (*e.g.*, harmonic dis-

turbance) or when, for instance, the importance of inertia of the liquid is diminished on account of its viscosity (*e.g.*, the use of a viscous oil as ballast). In fact one finds experimentally that whereas one can almost dispense with the follow-up system when tanks are filled with viscous oil, its presence is very desirable (all other conditions being the same) when the water is used as ballast, and when the manifestation of its inertia (overshooting of the ballast) is of importance.

One can readily see the reason for this in the above definition of the follow-up action. When tanks are filled with the oil, for instance, as soon as the control is withdrawn or changed the moving fluid loses its kinetic energy (corresponding to the old state of control) more rapidly on account of its friction than water ballast under similar conditions.

Conversely, if the water ballast is controlled by instruments with a follow-up connection, the same results as with oil can be obtained with a considerably smaller power consumption in view of the fact that the follow-up will limit the controlling action thus preventing the useless power expenditure, while in case of oil operating without any follow-up, apparently, good results are produced at the expense of power absorbed by the viscosity of the liquid.

One can thus see that the problem of an adequate follow-up control is just as vital to the efficiency of a stabilizing plant as the primary stabilizing control. If the latter *anticipates* the development of the disturbance *in its beginning* the former determines *the ending of* the stabilizing action without which the primary control would inevitably cause an over-regulation just as bad as no control at all.

Viewed from this angle the correct relation between the primary (*i.e.*, instruments) and the secondary (*i.e.*, follow-up) controls is of great importance and the practical solution of the stabilization problem by gravity methods depends largely on this point.

For instance, in the first investigated case, $\left(w = a \frac{d\theta}{dt}\right)$, as previously explained, the primary control is derived from the constrained gyroscope. One sees at once that the secondary control must be connected with a water level recording device, such as floats, etc., in order to interlock correctly both primary and secondary controls.

In the second case, $\left(\frac{dw}{dt} = a\frac{d^2\theta}{dt^2}\right)$, the primary control is produced by an accelerometer, hence the secondary must be connected to an apparatus measuring rate of flow.

Although by increasing the order of the time derivative responsive instruments and circuits, the anticipatory features of the stabilizing control are improved, other difficulties appear so that it is doubtful whether, in practice, anything higher than the third derivative control $\left(\frac{d^3\theta}{dt^3}\right)$ may be needed.

First, by going to higher time derivatives in the control of the ballast, the latter is not very definitely distributed between the tanks and may creep slowly from one side to the other. This creeping is generally limited by the ship itself which, on the average, has normal upright position by virtue of its stability. In addition a by-pass can be provided to eliminate from time to time the unbalanced condition of the tanks. The need for this is seen in the fact that in any process of integration a cumulative error is unavoidable and in case of a multiple integration a power series of time is associated with the constants of integration.

Another limitation is the design of instruments; for the first derivative control $\left(\frac{d\theta}{dt}\right)$ the small constrained gyro is probably the most suitable instrument.

For the second derivative $\left(\frac{d^2\theta}{dt^2}\right)$ control an inertia instrument responsive to angular acceleration is a very simple and efficient instrument having no rotating parts and not requiring any supervision whatsoever.

To obtain a control responsive to the third derivative, $\left(\frac{d^3\theta}{dt^3}\right)$, one can use any form of mechanical differentiating device (*e.g.*, eddy currents, liquid) or, still better, electrical control by a circuit coupled inductively with an accelerometer control provided the latter is continuous. Very satisfactory results were obtained by this

method although no use of the third derivative control was made in the experiments above referred to (7).

4. CONDITIONS OF ROLL QUENCHING IN A CONFUSED SEAWAY.

Approximation of Froude's theory as to the harmonic nature of moments exerted by waves on the ship ceases to be applicable in case of a confused erratic seaway. These conditions will be analyzed in the following.

It is somewhat misleading to judge the degree of regularity (or irregularity) of a given seaway by the rolling curve of the ship, because this degree of regularity depends on the properties of the ship perhaps just as much as on the character of the seaway.

For example, a vessel with high metacentric height will reflect in its rolling curve the character of the seaway to a greater degree than a vessel with a small metacentric height which will roll more in her own period and have a more or less regular rolling even in a comparatively confused sea.

On the other hand a raft, representing a limiting case of a high G.M. vessel, will be practically devoid of its own period and will reproduce in its rolling curve all the irregularities of the sea.

Conditions of the sea, on the other hand, may vary within wide limits. Regular trochoidal waves from time to time are observed under special conditions of the coast and weather. The Bay of Biscay, for example, is well known for its " quiet sea rolling " caused by a hardly perceptible long period ground swell of remarkable regularity, especially when the wind dies down.

Under other conditions, especially after a change in direction of the prevailing wind, owing to complicated conditions of interference of two patterns of waves from different directions, the seaway may become very erratic.

Large amplitudes or rolling are generally attained on ships of a more or less normal design (G.M. of the order of 2 feet, possibly 3.0 feet) in a series of rolls produced by a train of waves synchronizing approximately with the free period of the ship. In this manner, an approximate condition of regularity, when such a train is passing, seems to be necessary for attainment of large angles of rolling.

Erratic rolling appears, generally, past the envelope of these large amplitudes after a synchronous train has passed, and before the

next one begins its cumulative action on the ship. Reference is made to curve θ of Figure 2 representing an actual rolling record illustrating this particular condition.

Froude's theory of rolling among waves cannot be for that reason used with the harmonic second member in equation and a generalization is necessary.

The question arises as to how the action of stabilizing control of the type previously studied will be felt in case of an erratic sea (22).

Clearly this question cannot be answered in case of a discontinuous control, acting from time to time because in such a case the cause-effect relation will not hold in instantaneous values and the control of stabilization may become not less erratic than the sea. This is perhaps one of the reasons for a common belief that all stabilizers are equally useless if called upon to quench an erratic sea.

Conditions, however, are radically different in case of a continuous control when the cause-effect relation is fulfilled by continuity. It is easy to show that in such a case the operation of the stabilizing control is capable of reducing the amplitudes whatever the character of erratic impulses might be, provided the power plant has a sufficient capacity to be able to impart on the ballast necessary accelerations in the required amounts corresponding to the instantaneous state of rolling.

Let $f(t)$ represent the disturbing moment of the wave slope considered as a certain unknown function of time. It is obvious that any attempt to approximate actual complicated conditions by a known function are meaningless in this case, because, the subsequent waves differ from each other in form, period and wave length and the problem reduces itself to an investigation of the action of a series of isolated impulses following each other more or less erratically. The problem under these conditions is entirely indeterminate from the standpoint of a quantitative solution. It is possible however to from an idea about the effect of the stabilizing apparatus if each individual impulse is considered separately.

The problem to a certain extent is similar to that of transmission on telephonic lines acted upon by isolated electromagnetic impulses.

It is known that the general solution of differential equation, in such a case, is represented by a Fourier integral (23):

$$\theta(t) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{+\infty} \varphi(\omega) \sqrt{r(\omega)}\, e^{j\omega(t-\phi)}\, d\omega \quad . \quad . \quad (26)$$

where $\varphi(\omega) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{+\infty} f(t)\, e^{-j\omega t}\, dt$ and . . . (27)

$$f(t) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{+\infty} \varphi(\omega)\, e^{j\omega t}\, d\omega. \quad . \quad . \quad . \quad . \quad (28)$$

The function $\sqrt{r(\omega)} = \frac{1}{\sqrt{(b^2 - \omega^2)^2 + a^2\omega^2}}$

is again the *resonance function* corresponding to the left member of Froude's differential equation, although the expression " resonance " has no particular meaning in this case when the action of isolated impulses is considered.

Equation (26) shows, however, that the stabilizing action is felt also in this case by the decreased value of the resonance function which, all other things being equal, reduces the value of the amplitude θ (equation (26)) corresponding to the passage of the impulse $f(t)$.

We reach thus an important conclusion for the following, *viz.:* (1) *if the stabilizing control is continuous and (2) if the stabilizing torque produced in this manner is in phase with angular velocity; the stabilizing action thus obtained is effective both for regular and for erratic rolling.*

During the former this stabilizing action admits of a predetermination on the basis of Froude's theory.

During the latter, as long as the nature of disturbing impulses remains unknown quantitatively, no such predetermination is possible although the quenching action of the stabilizer is contained implicitly in the decrease of the resonance function $r(\omega)$ which accounts for the corresponding decrease of the stabilized erratic roll as per equation (26).

If, however, experimental data relative to $f(t)$ is available, predetermination of the stabilizing action becomes possible even in this case through the instrumentality of the Fourier integral which can then be calculated. From the preceding study it follows that the difference between the damping and compensating action of a stabilizer vanishes at a closer study. If the control is arranged to re-

spond not to the angular motion of the ship but, for instance, to pressures set up by waves on different parts of the ship, to anticipate the disturbance in this manner, nothing is changed in the preceding line of argument. Such method of " outboard control " would not lead to anything new as compared to what has been already studied; because the external moment of waves measured in this way is inevitably accompanied by corresponding angular acceleration and one falls back, exactly, to the same methods of control when these accelerations are measured by an accelerometer instead of the more or less round about, " outboard " methods.

An adequate stabilizing control by gravity methods consists in absorbing the work of the wave slope in raising an extra weight at all times and the whole problem is to shift this weight so that it will follow continuously the instantaneously rising side of the ship. The question of the damping or of the compensating action which a stabilizer may produce thus represents two different ways of looking at the same dynamical fact, namely, absorption of the extra energy of the waves by a continuous raising of the weight throughout the rolling whether regular or erratic. Analytically the first viewpoint is expressed by equation (13) and the second by equation (17) but these two equations are equivalent.

It is of importance to note that the continuity of stabilizing control seems to be more important, at least in principle, than the particular method of timing the stabilizing moment. One could readily conceive, at least theoretically, an arrangement in which the control, while still being continuous, would be proportional to the square of angular velocity corresponding to an effective increase of coefficient k_2 in the differential equation (1) of Froude's theory. No particular practical advantage would be gained however in this case; on the contrary it would be more difficult to secure an accurate stabilization (that is for small $\frac{d\theta}{dt}$) because, for small oscillations, a control of this kind would be less sensitive and the amplitude would go up to larger values at which $\left(\frac{d\theta}{dt}\right)^2$ becomes appreciable and the control begins to function. However, even in this case the continuity of control will secure the stabilizing action both for regu-

lar as well as for erratic sea although the predetermination of quenching would become rather difficult because the equation of rolling would not be linear.

If the control is not continuous (*e.g.*, involves the usual relays, contactors, or similar devices operating from time to time) the instantaneous balance between the cause (*i.e.*, the disturbing wave slope) and the effect (*i.e.*, stabilizing moment), as already mentioned, ceases to exist and the efficiency of roll quenching falls off very rapidly, especially in an erratic seaway, when the cause-effect relation between the ship and the stabilizer may disappear altogether, and the stabilizer may become inoperative.

It must be borne in mind that one second of time lost on account of discontinuous action of a certain contactor for a vessel having a period of $T = 12$ seconds is equivalent to a phase angle of 30 degrees in the action of the stabilizing control and, if the sea is erratic, this may be more than that necessary to cause the equipment to drop out of step with the disturbance and have little chance to regain it again as long as the seaway remains erratic.

Viewed from this angle modern thermionic controlling devices and circuits afford almost an ideal instrumentality for handling complicated conditions arising from the very nature of the disturbances impressed on the ship in an erratic sea and they are capable of following these disturbances with practically perfect continuity.

5. EFFECT OF TIME LAGS IN THE TRANSMISSION SYSTEM.

In addition to the above conditions of (1) continuity and (2) proper time phase, the influence of time lags must be investigated.

The time lags can be considerably reduced if proper electrical circuits are used, but can never be totally eliminated. Mathematically, the effect of time lags can be taken into account by observing that, whereas the dynamical conditions (action of the sea, etc.) relates to the instant t (the present instant), the stabilizing moment inevitably relates to the retarded time $t - \tau$ (the past instant) where τ is the time lag.

Hence the stabilizing moment can be represented:

$$S = K_s \frac{d\theta}{dt}(t - \tau) = K_s \frac{d\theta}{dt} - K_s \tau \frac{d^2\theta}{dt^2} \quad . \quad (29)$$

where $\frac{d\theta}{dt}$ $(t - \tau)$ designates the value of $\frac{d\theta}{dt}$ at the retarded time $(t - \tau)$. Equation (23) represents Taylor's expansion applied to S considering τ as small in comparison with ship's period as well as with the period of the sea and neglecting terms with τ^2, τ^3, etc. Substituting (29) into equation (3) the latter will not be changed except the first term which instead of I will contain $I' = I - K_s$; this will be also applicable to the constants a and b^2 of the transformed equation (4) namely $a = \frac{K}{I^1}$, and $b^2 \frac{Wh}{I^1}$.

If time lag is large enough more terms must be taken in expansion (29) and this raises the order of differential equation on account of the terms with $\frac{d^3\theta}{dt^3}$ $\frac{d^4\theta}{dt^4}$ etc.

The " still water " rolling of the ship which generally determines the decrement of free oscillations may become erratic in this case. The stability conditions are not easy to predetermine unless one goes through rather laborious methods of predetermination based on the applications of the, so called, Hurwitz theorem (24); roll quenching efficiency generally decreases especially in an erratic sea when the time lags become too great.

One should not, however, be too pessimistic with predetermination of the order of magnitude of the time lag which can be tolerated without an appreciable loss of roll-quenching efficiency. For a small vessel of destroyer type the free period of the ship is of the order of 10 seconds. With a time lag of 1 second, the second term in Taylor's expansion is of the order of 5 per cent and can thus be neglected. On the other hand, the time lag in an installation of about 30 H.P. corresponding to the vessel of this type can be made less than 1 second if, for example, inductive circuits are controlled by vacuum tubes which reduce their time constant and thus decrease time lags very appreciably. On a large liner with $T = 25$ seconds or so, for example, a time lag of 1 second is altogether negligible, the error arising from the omission of the second term in Taylor's expansion being of the order of 10^{-2} only.

6. CONCLUSIONS.

The preceding analysis permits of establishing a series of important conclusions relative to the stabilizing control:

A. *Continuity of controlling action.* It was shown that, in case the control is continuous, the stabilization is effective even when the the disturbances are erratic, and when, consequently, Froude's theory ceases to be applicable. Experimental investigation on a model confirms entirely this theoretical conclusion (25). The continuity of controlling action for active tank stabilizers is more important than for gyroscopic stabilizers. In fact, in the latter, the control of precession is discontinuous insofar that the control gyro reverses the precession motor's starting contacts in a discontinuous manner on the zero points of angular velocity. Between the two subsequent reversals, the gyro is precessed with a substantially uniform velocity. The stabilizing component of the gyroscopic torque acting in the plane of rolling varies, however, continuously as the cosine of the angle of tilt, so that in spite of a discontinuous control of precession, the gyroscopic torque acting in the rolling degree of freedom varies as a sinusoidal function of time and is thus capable of compensation on the average of the fundamental harmonique of disturbance. A similar condition (*i.e.,* discontinuous control) in case of the water ballast would be far less satisfactory insofar as in this case the rate of flow, considered as a function of time, would approach rectangular form, which, aside from purely hydrodynamical difficulties, would lower the roll-quenching effect very appreciably because the cause and the effect would not match each other properly in this case.

These conclusions are applicable only to the fundamental harmonic of rolling motion, subject to the analysis by Froude's theory. Conditions in reality are still less favorable for a discontinuous control if the sea is erratic. For example, in case of a vessel of 25,000 tons, having G.M.—3′0 and subjected to action of a synchronous wave slope of 5 degrees, with period of 18 seconds—the maximum variation of the disturbing moment acting on the ship during one second is of the order of 2500 foot tons. If the control is even partially discontinuous, *e.g.,* by a certain contactor operating from time to time (say every second) and the sea is erratic, the time interval of one second will constitute a zone of insensitivity for the

control during which the cause-effect relation between the sea and the stabilizer may be lost. When it is again regained at the following contact, in addition to what the stabilizer will have to furnish at that time, there will be a balance of about 2500 foot tons left from the preceding period when it was idle during one second, and this balance may vary erratically (*i.e.*, may be either added or subtracted) if the seaway is erratic enough. The above elementary example shows, in a non-mathematical manner, what has been ascertained already in section 4 analytically, namely, the importance of a perfectly continuous controlling action, capable of following the disturbance by continuity.

B. *Anticipatory features of stabilizing control.* If the sea at all times were of a regular trochoidal pattern, the problem of anti-rolling stabilization would be an easy one. Being given, however, the fact that the sea is never regular, but very frequently has a pronounced erratic character, it is not sufficient to follow the disturbance by continuity but becomes necessary to use more refined means of detection. The controlling system must be provided, for that reason, with phase advancing devices which will contribute to roll quenching efficiency on the one hand and reduce the wear and tear on the power plant on the other; as was shown, these anticipatory features of detection are closely related to the use of higher time derivatives responsive devices in the control system.

C. *Time Lags.* Even if one assumes that the preceding two conditions are fulfilled, but the controlling action is steadily delayed somehow (*e.g.*, necessity to reverse certain rotating machines, build up magnetizing current in an inductive circuit, etc.) the roll quenching efficiency will inevitably be reduced. Suitable methods should therefore be applied (*e.g.*, control of inductive circuits by high resistance devices, such as vacuum tubes, use of movable blade impeller pumps, eliminating thus the necessity for reversing rotating machinery, etc.) in order to insure the rapidity of controlling actions.

D. *Correct relation between the primary and the secondary controlling systems.* It was seen that for a given ship the roll-quenching efficiency depends to a large extent on a correct relation between the intensities of the primary control of detection and the secondary or follow-up control. If the former predominates, complicated phenomena of over-regulation (reference (7) Figure 7) may lower

the roll-quenching efficiency. If, however, the latter is made too strong, the loss of efficiency may be such that the stabilizing moment will have no chance to develop to its full extent.

7. OPINION.

One can thus readily see the reasons why the active tank system for so many years remained in a background without any practical success in spite of all its potential possibilities. The main reason for this lies, in our opinion, in the difficulty of producing a control system possessing all of the above enumerated features. Only recently, with the advent of new controlling devices, developed in other connections, it became possible to solve the problem of handling correctly the displacements of huge masses of ballast to match instantaneously ever-changing disturbing moments which an erratic seaway applies to the ship.

One has thus all reason to believe that after almost fifty years of strenuous effort toward its practical solution, the problem of anti-rolling stabilization has a great chance to find a simple and efficient solution in the active tank system.

BIBLIOGRAPHY.

(1) An excellent review of the prior art is given in a series of editorial articles published in the "The Engineer" (London) in 1930.
(2) Transactions I. N. A. Volume XXIV.
(3) U. S. Patent No. 1,232,619.
(4) U. S. Patent No. 1,533,328.
(5) Transactions I. N. A., Volume XXXIII.
(6) U. S. Patent No. 1,708,679.
(6') "The Engineer," 1934.
(7) N. Minorsky, "The Engineer," August 17, 1934.
(8) Transactions I. N. A., Vol. XIV. Also Sir Ph. Watts, "Shipbuilding," Enc. Britt.
(9) Sir Ph. Watts, loc. cit.
(10) Pollard et Dudebout, Théorie du Navire, Chapt. XXXIX.
(11) E. L. Attwood, Theoretical Naval Architecture, 1922, p. 409.
(12) H. Frahm Schiffbau Techn. Gesell. 12 (1911), page 351.
G. Bauer Der Schiffmaschinenbau Bd. 2, page 112, 1927.
E. Hahnkamm, Werft Red. Hafen 13, 1932.

(13) E. Hahnkamm, Loc. cit., also Zeitsch, fur. Angew. Math und Mec., June, 1933.

(14) A. Pommellet, Considerations sur le roulis, Assoc. Technique, Maritime, May, 1932.

(15) Between coefficients k_1 k_2 of damping, and corresponding coefficients a and b of decremental equation there exist the following relations (Attwood, loc. cit. page 392) : $a = \frac{1}{2} k_1 \frac{\pi^2}{WhT}$ and $b = 4/3\ k_2 \frac{\pi^2}{WhT^2}$ where T in Attwood's notations is *one half* the period of the ship.

(16) For a more complete study of the problem, in addition to useful gravity reaction of the stabilizer on the ship, a secondary kinetic reaction should be considered. The solution given by equation (3) is for that reason only approximate corresponding to an already established state of stabilization from time to time disturbed by transients. As another correction, variation of apparent gravity, both in magnitude and direction should be taken into account. This correction, generally, is small enough from the standpoint of the first approximation with which we are concerned here and will not be introduced.

(17) Attwood, Loc. cit. p. 400.

(18) Attwood, Loc. cit. p. 388.

(19) N. Minorsky, "The Engineer," August, 1934, more specifically Fig. 7.

(20) Sir Ph. Watts, "Shipbuilding," Enc. Britt.

(21) It must be noted that the residual quenched rolling generally has a different phase and shape than the corresponding non-quenched rolling. In this section we are not concerned with the dynamics of ship stabilization but with the problem of control of the ballast *for a given* residual rolling. Example of Fig. 2 must be considered from this last mentioned viewpoint.

(22) In case of a continuous control in accordance with equation (24) used in anti-rolling experiments on a model, it was found that the stabilizing action is very satisfactory even when the disturbance is entirely erratic. (See "Engineer," Loc. cit., Fig. 8.)

(23) Courant Hilbert, Methoden der Mathematischen Physik, 1924, p. 227.

(24) Hurwitz has given (Math. Annalen t XLV) general conditions of stability of a system whose motion, or state in general, is governed by a linear differential equation, in the form of a series of equations between the subsequent, so-called, Hurwitz determinants (or principal minors) derived from the characteristic equation $a_0x^n + a_1x^{n-1} + \ldots + a_{n-2}x^2 + a_{n-1}x + a_n = 0$ corresponding to the given differential equation in the following manner:

$$H_n = \begin{vmatrix} a_1 & a_0 & 0 & 0 & 0 & 0 & \cdot\cdot \\ a_3 & a_2 & a_1 & a_0 & 0 & 0 & \cdot\cdot \\ a_5 & a_4 & a_3 & a_2 & a_1 & 0 & \cdot\cdot \end{vmatrix}$$

where Hurwitz determinants (or principal minors) are indicated by dotted lines. Hurwitz has shown that the necessary and sufficient conditions for stable roots (*i.e.*, whose real part is essentially negative) is that n first principal minors be positive. A system thus satisfying n Hurwitzian conditions is always stable in the sense of the method of small oscillations. See also in this connection A. Blondel, Journal de Physique, April, 1919.

(25) N. Minorsky, Loc. cit. Fig. 8.

NOTICE.

AMERICAN SOCIETY OF CIVIL ENGINEERS

33 WEST THIRTY-NINTH STREET

NEW YORK CITY

NEW YORK, *January 2, 1935.*

BULLETIN ON

SURVEY OF THE ENGINEERING PROFESSION.

There will go forward in early 1935 a Survey of the Engineering Profession which undoubtedly is the greatest opportunity ever offered to study that profession from top to bottom and from side to side. It is hoped that the survey will reach every man who thinks of himself as an engineer, from sub-professional man and recent graduate up to chief engineers and consultants, and laterally from civil, mechanical, electrical, mining, chemical—through all the long list of branches and types of engineers. It will be a country-wide study, with the basic information coming directly from individual questionnaires.

Much speculation has been attempted as to the effect of these last few troubled years on engineers. The intent of the Survey is to replace such speculation with facts. Engineering surveys are constantly being applied to other problems—now one is to be made to help in planning intelligently for the future of the profession itself.

The objects of the survey are to determine the factors which contribute to tenure of employment and compensation, degree of unemployment and sources of re-employment, the influence of education, the relationship of engineering to other professions, etc. Analyses will be made by the Bureau of Labor Statistics and published as a Federal document.

Many factors may affect an engineer's employment, especially in times such as the present. Education, years of experience, type of work, where the man practiced, whether previous employment has been continuously in one organization or in one type of work or scattered among several, membership in professional or technical societies, registration (in States that require it), salaries received in the past, typical duties, special qualifications—all these and many others may have a definite bearing on his present situation. In addition, it will be helpful to have other data for the whole group, such as how many are employed and how many are unemployed, how many have been or are now on some form of work relief, what salaries are being paid today under emergency conditions as well as under so-called normal employment, and so on.

The questionnaire is being prepared by Dr. Lubin, of the Bureau of Labor Statistics of the U. S. Department of Labor, with the aid of members of several engineering societies, selected for their experience in employment studies.

The Bureau of Labor Statistics will print the questionnaire, mail it, receive the returns, analyze them and publish the results. It has splendid clerical and

mechanical facilities, and its use of the franking privilege will eliminate a very large direct expenditure for postage.

One thing the Bureau can not do is to prepare the mailing lists. American Engineering Council has undertaken to do this. What is necessary is a file of names and addresses, stencilled or typed on 3 x 5 cards, sorted by states and arranged alphabetically, with duplicates laid aside. It will not be possible to utilize printed lists, such as rosters or year-books, nor are there facilities available for transferring such information to cards. When it is understood that there may be more than 150,000 cards to sort, it will be obvious why the cards themselves must be furnished. Application has been made to F. E. R. A. for assistants to do the sorting of the cards. The American Society of Civil Engineers will furnish space for these assistants to work and will supply at its expense all necessary correspondence, postage and incidental facilities and equipment.

The method of securing the card lists is as follows: requests were sent to 263 known engineering societies, associations and clubs, to 32 state registration boards, and to 156 engineering schools (the latter with respect to their graduates of the last five years only), asking that the cards be prepared and sent to George T. Seabury, Secretary of the American Society of Civil Engineers, 33 West 39th Street, New York City. These units were also asked to utilize their available facilities to urge their members to fill out the questionnaire carefully, earnestly, and to the last detail, and return it promptly to Washington. The questionnaires will be mailed out early in the year.

The 1930 Census indicates that 226,136 persons in the United States consider themselves engineers. At the present moment there have been received cards from 148 groups totalling perhaps 135,000 names. Of course this number includes duplicates. Such prompt response is extremely gratifying. Every mail brings more cards and letters saying that cards are being prepared or are on their way. In most cases it has been a real task and frequently irksome for the collaborating unit to put its membership data on cards. The cooperation, however, has been splendid, and very few indeed have reported that they are unable to help—and then usually because of financial hindrance or a technical handicap.

Even if all the associations, boards, clubs and schools should supply their membership lists, there will still remain some engineers who do not belong to any such organizations. Since the hope is to send the questionnaire to every engineer in the country, or at least to a very extensive sampling, it will help if any person who knows of an unaffiliated engineer will send in his name and address on a 3 x 5 card so that it may be placed in the file. As another example of needed names there should be mentioned those former members of organizations who have had to drop out within the last few years for financial reasons. Their names are much wanted in order that questionnaires may surely get to the men perhaps the hardest hit. Organizations may be very helpful in this respect.

The questionnaires will be absolutely confidential. The answers will be punched on tabulating cards and fed through sorting machines. No person connected with the work is interested in the individual returns. It is certain, however, that large numbers of engineers will be interested in the results. Each individual engineer will realize also that these results will be incomplete if HE does not send in his own report, completely filled out, and PROMPTLY.

American Engineering Council's Committee on Engineering and Allied Technical Professions sponsors the program, and all associations of engineers and all individual engineers are asked to participate enthusiastically.

NOTES.

INDEX TO NOTES.

PROPELLER CAVITATION STUDIES.
Institution of Naval Architects, July 11, 1934.

SAVING LIVES AT SEA.
By Kenneth Porter, Member.

PREVENTABLE FIRE AT SEA.
Scientific American, January 18, 1935.

SAFETY IN MARINE TRANSPORTATION.
Technical News Bulletin, National Bureau of Standards, December, 1934.

COMPARATIVE COST OF MOTOR SHIPS AND STEAMSHIPS.
Journal de la Marine Marchande, December, 1934.

ELECTRIC PROPULSION SYSTEMS.
Shipbuilding and Shipping Record, December 6, 1934.

THE BIROTOR FIXED—VANE ROTARY PUMP.
Engineering, December 7, 1934.

WEAR TESTING MACHINE.
The Engineer, December 7, 1934.

LUBRICATION.
Power, Mid-December, 1934.

BOILER FEED WATER.
Power, Mid-December, 1934.

WELDING VS. CASTING.
Power, Mid-December, 1934.

WASTED EFFORT IN THE PACIFIC.
The Aeroplane, December 12, 1934.

DIESEL PROGRESS.
Power, Mid-December, 1934.

AERONAUTICAL ENGINEERING.
The Aeroplane, December 5, 1934.

A NON-FERROUS ALLOY.
The Aeroplane, December 5, 1934.

NORTH ATLANTIC CATAPAULTS.
The Aeroplane, December 5, 1934.

PROPELLER CAVITATION STUDIES.*

LIEUTENANT-COMMANDER C. O. KELL (C.C.), U.S.N., EXPERIMENTAL MODEL BASIN, WASHINGTON, D. C.

ABSTRACT.

Based upon the information that has been obtained from experiments conducted in the variable pressure tunnel at Washington during the past two years, the different forms of propeller cavitation are discussed. The characteristics of face cavitation and two forms of back cavitation are described. Methods of correction as far as are known are suggested.

Tests to determine the effect of gas in solution in water on propeller cavitation are described, and the results are given.

Finally, the calculated lift coefficients for typical propellers at the apparent inception of back cavitation are plotted against cavitation indices, in an effort to obtain a criterion by which back cavitation on propellers may be avoided in design work. These data are compared with a theoretical relation between the lift coefficient and cavitation index, which has been offered by Lerbs as a criterion.

INTRODUCTION.

Two years ago Captain E. F. Eggert (C.C.), U.S.N., presented a paper (1)† before the American Society of Naval Architects and Marine Engineers on the subject of "Propeller Cavitation." In this paper he discussed the limitations of criteria that had been proposed prior to that time for predicting the speed at which cavitation in marine propellers would occur. In place of these earlier criteria he offered a formula by which we can determine the approximate speed at which back cavitation becomes serious for any ordinary type of submerged propellers. This formula was based upon the results of research work done on the subject until that time in the variable pressure water tunnel at the Experimental Model Basin in Washington.

Since the publication of Captain Eggert's work it has been my privilege to have been associated with the work at the tunnel. This occasion affords a most acceptable opportunity for me to record briefly the developments in the study of cavitation at the Washington tunnel during the past two years, and to benefit from discussion with those who have been engaged on similar work.

Since the installation of the Washington tunnel, work has been constantly carried on to perfect the methods of testing and the development of more efficient equipment for making more accurate measurements. We are aware of the limitations of our present methods of testing, and we do not hold that our results at present are not subject to correction. We are in need of more results from ship trials with which to compare our tunnel tests before we can feel certain that our test methods are such that the results check satisfactorily with the ship performances. We realize that there yet remains much to be accomplished in this direction, but we do believe that the changes that have been made and the work that has been done have resulted in progress in the study of cavitation and the building of better propellers. I refer to these developments because for the most part they are responsible for our obtaining more accurate data upon which to base our studies. The most important of these developments are the more reliable measurements of the

* Read at the Summer Meetings of the Seventy-fifth Session of the Institution of Naval Architects, July 11, 1934, SIR WILLIAM BERRY, K.C.B., Vice-President, in the Chair.

† See bibliography at end of paper.

speed of water past the propeller in the tunnel, more dependable torque records, and more accurate pressure measurements within the tunnel. The present measurements of water speeds in the tunnel make possible closer approximations of the true slip ratios that are obtained in self-propelled tests. These are vital in obtaining the correct characteristics in the development of cavitation on the test propellers. In an effort to check the accuracy of the representations of cavitation obtained in the tunnel, an observation port has been built in the shell of one of the United States destroyers. From this we hope to study the development of cavitation on propellers of different designs. Another addition which has proved most helpful in our studies is a stroboscopic light, which has made possible quite satisfactory continuous observations of propellers under test. In this way greater accuracy is obtained in determining the speeds of advance at which cavitation becomes apparent, and the manner of inception of cavitation and its development through the range of higher speeds can be studied. It will be recalled from an earlier description of the water tunnel at Washington (2) that the tunnel is of the open jet type, which makes possible the illumination of the propeller chamber by exterior light.

DISCUSSION OF FORMS OF CAVITATION.

In the past two years a great many propellers have been tested. The different types of cavitation have been clearly and consistently defined. These are face cavitation and back cavitation in its two forms. I should like to briefly discuss each of these forms of the phenomenon, but before doing so it is perhaps well to describe the methods of testing used when these data on the beginning and the developments of cavitation were obtained.

In all standard tests in the tunnel the procedure in model basin self-propelled tests is followed. From the data obtained in the self-propelled tests in the basin the thrusts at the different speeds are calculated from the E.H.P. curves, corrected for the corresponding thrust deduction factors. The correct speeds of advance are also determined, using the wake fraction values obtained in the self-propelled tests. In the tunnel tests the proper pressure according to scale is maintained constant throughout the tests. At each speed the corresponding speed of advance is maintained as constant as possible in the tunnel. The corresponding thrusts are maintained on the dynamometer by a minimum of regulation of the revolutions. Each observation is made for the duration of one minute, and the data are required to be repeated before being accepted. Some slight regulation in revolutions is found necessary in the test interval to maintain a given thrust. This is due to the fluctuation of voltages of the power used at the tunnel. Power readings and revolutions for each observation interval are read from the dynamometer. The advantage of this form of test is that, if correctly done, it gives the sequence of developments of cavitation as they take place on the ship. The results of these self-propelled tests in the water tunnel agree closely with those made in the basin up to speeds at which the efficiencies of the propellers become affected by cavitation.

Face cavitation, if it be present, is found to exist at the lower or throughout the so-called cruising speeds, when minimum slip ratios occur. If the slip be low, the blade sections abnormally thick with sharp leading edges, and with blades close together as in the case of four-bladed propellers, face cavitation will probably exist. If so, it may continue throughout quite a range of speeds, depending upon the increase in the slip ratios with increase in speed. Face cavitation generally sets in at the root sections. This may be accounted for as being due to the effect of blade interference and to the thicker sections at the base of the blades.

Perhaps the most important fact about face cavitation which our data have established is that its presence does not have any effect upon the efficiency of the propellers that we have been able to measure. A great many model propellers which we have tested have been found to be in face cavitation, and in no case when face cavitation has been observed have either the revolution or the power curves in self-propelled tests been affected by it. I believe that our data have answered the question once and for all as to whether or not face cavitation has any appreciable effect on the thrust of a propeller. The fact that it does not can be understood when we remember that it is the back of the propeller blade and not the face that governs in propeller performance.

No doubt there are those who will find it difficult to accept these results as conclusive on the question. They have witnessed instances in which the power of a propeller was affected by cavitation, and the propeller was known to be working in face cavitation from the markings on the blades. In explanation of such cases, it is probable that face cavitation did exist at the lower speeds, and that back cavitation set in at higher speeds, and was the cause of the loss of power. In fact, we have frequently found that both face cavitation and a form of back cavitation exist simultaneously through a short range of speeds. In such cases, as the back cavitation develops and causes the slip ratios to increase by causing the efficiency of the propeller to fall off, the face cavitation decreases, until finally it disappears. When this occurs there is no change in the slopes of the power and revolution curves which would indicate any effect of face cavitation.

What I have said thus far must not be taken to infer that we have concluded that face cavitation is not objectionable and need not be avoided. We know that it is a cause of erosion on the blade face, and we suspect that it may be the source of objectionable and even serious vibrations. We hope to be able to answer this question after full-scale tests.

For these reasons it is fortunate that face cavitation can be easily corrected and avoided. If the propellers be operating with a reasonable slip ratio—that is, not much less than that corresponding to maximum efficiency—to correct face cavitation it usually has been only necessary to round the leading edges of the root sections of the blade. At most, it has been necessary to set back the face of the blade, beginning a short distance back from the leading edge. Because of the shape at leading edges, propeller designs having aerofoil sections do not show face cavitation when operating at or near normal slip.

While face cavitation is not believed to be very serious, in that it can be avoided in properly designed propellers, back cavitation is quite a different matter. Back cavitation in definite stages of development does have a tremendous effect on propeller performance. In discussing back cavitation I wish to consider separately the two forms of it which we have observed at Washington, because the two forms have different characteristics in their inception and development, and they affect propeller performance differently. The first of these forms of the phenomenon is that which first appears as vortices off the blade tips, and develops into leading edge cavitation with increase in tip speed and thrust. The other form of back cavitation is that which first appears as bubbles striking on the following part of the blade back, with reference to the maximum thickness of the blade sections. To distinguish this form of the phenomenon from leading edge cavitation, it will be referred to as "burbling." The word is taken from the name of the analogous phenomenon in aerodynamics.

Consider, first, back cavitation at the leading edge of the propeller blades. It is known to occur at the leading edge and at the blade tips because the lowest pressures on the blade back exist there. With the tip sections of the

propeller operating at a high slip, the flow of water about the leading edge of the sections comes from the pressure side of the blade, and at high tip speeds it is unable to round the leading edges of the tip sections and follow the contour of the back of these sections. The rate of development of the phenomenon is dependent upon the shape of the tip sections of the blade, in particular the form of the leading edge, the angle of attack, and the tip speeds. Leading edge cavitation first appears as vortices passing off the blade tips, which form helices in the propeller wake. With each increase in speed and thrust, the pressure in the tunnel remaining constant, there is a corresponding development of these vortices until what appears to be an independent flow is formed across the blade tips, and bubbles appear down along the leading edge of the blade. As the speed is further increased, the width of the flow across the blade tips increases, and the trailing edge of this flow passes off into the propeller wake as wider bands of vortices. The cavitation along the leading edge becomes wider, and finally, when well developed, extends across the entire back of the blade as a part of the independent flow which began at the tip sections. This flow remains intact until it is well clear of the following edge of the blade.

The development of leading edge cavitation on propellers is very similar to that described by Walchner (3) and Ackeret (4) in their tests with hydrofoils having sharp leading edges, except in one particular. Walchner found that with the development of cavitation on a hydrofoil, set at a given angle of attack, the origin of cavitation moved from the leading edge of the foil to the section of maximum thickness with decrease in the cavitation index. This, he explains, is due to the shifting of the stagnation point with the beginning of cavitation from the pressure side of the foil to a position in front of the leading edge. No similar phenomenon has been observed in our model propeller tests. It is true that "burbling" has been found to set in on the following half of the back of the blades, in some instances while leading edge cavitation existed. However, the inception of "burbling" did not decrease the intensity of leading edge cavitation in any case. It is believed that the explanation of this is that with propellers designed for ships of ordinary types, any change in the angle of attack after cavitation has set in is an increase, which tends to aggravate the conditions that cause leading edge cavitation.

Interesting facts about leading edge cavitation are that it can exist on the wider blades through a wide range of speeds without affecting the efficiency of the propeller, but after the efficiency begins to fall off, progress toward general breakdown of the propeller action is rapid.

It has been found possible to delay the beginning of leading edge cavitation, and in some cases to confine it to tip vortices, by methods which involve some sacrifice in efficiency. It has not been possible to do much towards its correction by altering the form of the thin blade sections at the tips. It can be most effectively delayed or reduced by decreasing the angle of attack at the outer sections of the propeller blades. This can be accomplished by increasing the propeller diameter and accepting a lower pitch ratio with consequent lower efficiency, or by twisting the blade to obtain lower pitch ratios at the outer blade sections.

The second type of back cavitation, which will be called "burbling" in this discussion, is that which forms on the trailing part of the blade, speaking with reference to the maximum thickness of the sections. It is the most interesting form because it presents the most difficult problems for solution.

In the tests of model propellers it has not been possible, thus far, to positively determine the location of the origin of this phenomenon. Our experiments with hydrofoils indicate that the origin is at the maximum thickness

of the blade sections. This was also indicated by Walchner (3) and Ackeret (4) in their tests of hydrofoils. The characteristics of the phenomenon in flow and extent as described by Walchner and Ackeret correspond with our observations on the model propellers only after the "burbling" type of cavitation has been fully developed, and long after the efficiency of the propellers has been affected by it. It is interesting to note here that Sir Charles Parsons succeeded in photographing the different forms of cavitation on propeller models. These records were published by Mr. S. S. Cook (5). His record of the "burbling" type of cavitation on the blade back is of particular interest because it appears to be identical to that which we have so often observed in the Washington tunnel.

At the first appearance of "burbling" on the propeller model we see intermittent impacts of bubbles on the following half of the back of the blade. It is believed these are condensation impacts, and occur when the bubbles of cavitation reach the zones of higher pressure and collapse with some force on the blade back. The first of these bubbles is most generally seen near the trailing edge of the blade in a definite small area. If the root sections be abnormally thick, "burbling" usually appears first near those sections. It has never been observed nearer the tip sections than about seven-tenths of the radius from the center of the propeller. With decrease of the cavitation index $\frac{p_a}{q}$,* accomplished by increase in speed, with pressure constant, the area affected is found to expand both radially and across the width of the blade towards the maximum thickness of the blade. When completely developed, most of the following part of the back of the blade is affected and the trailing edge of the combined cavity formation is clear of the blade. At this stage it corresponds to the description given by Walchner (3) and Ackeret (4). But even in this stage of development the cavitation is confined to the following half of the blade. This was investigated by an interesting series of propellers which were similar in all respects except as to the location of the maximum thickness in the width. In one of these propellers the maximum thickness was located at one-third of the chord length back of the leading edge; in a second one the maximum thickness was in the middle of the chord length as in the typical ogival section. In the third propeller of the series, the maximum thickness was at two-thirds of the chord length from the leading edge. The three propellers were given identical tests. The first propeller, with the longer slope from the maximum thickness to the trailing edge, broke in efficiency at the lowest speed, and its rate of decrease in efficiency was more rapid than in the other two propellers. The third propeller, with the greatest slope in the following half of the blade back, showed the greatest resistance to the effect of cavitation. This perhaps is due to the smaller percentage of the area of the back being affected in the third propeller.

It has been found in some tests that as the propeller efficiency falls off due to "burbling," with the resulting increase in slip, leading edge cavitation may also develop. In such cases, leading edge cavitation tends, with the "burbling" type, to lower the speed at which the complete breakdown of the propeller occurs.

The "burbling" type of cavitation is most sensitive to small changes in the cavitation index. This characteristic is responsible for what appears to

---

* p_a is the absolute pressure in the tunnel, $q = \frac{\rho}{2} v_r^2$, the dynamic pressure, in which ρ is the density of water and v_r the relative velocity of flow to the propeller blade sections.

be a condition of instability in performance of the propeller within cavitating range of speeds. In some cases this condition of apparent instability has made it impossible to accurately repeat test data within the range of speeds in which the "burbling" type of cavitation exists. This makes it very difficult to predict accurately the power required for propulsion after this type of cavitation has set in.

Experiments with model propellers of the destroyer type have been made that show that a difference in static head of 3 feet of water makes a very measurable difference in the cavitation index at which "burbling" occurs, and in the rate of its development. This is significant, in that it indicates the necessity of estimating as accurately as possible the depth of submergence of the propellers at the different speeds, when tunnel tests are used to predict the power of ships.

EFFECT OF DISSOLVED GASES IN WATER ON CAVITATION.

Tests have been made in the water tunnel to determine if the cavitation of propellers might be affected by gases dissolved in the water. Five model propellers were tested for cavitation in the tunnel in ordinary tap-water by the standard self-propelled methods. All tests were made twice under the same conditions for the purpose of checking results.

Upon completion of the tests in tap-water, carbon dioxide gas was bubbled through the water from the bottom of the tunnel from a cylinder containing the liquefied gas at the rate of about 15 pounds of gas an hour until a concentration of about 120-130 c.c. of gas per liter of water had been attained. This gas content was about twice the gas content of sea-water in the North Atlantic at a temperature of about 60 degrees F.

The self-propelled tests with the five model propellers were then repeated with the water saturated with carbon dioxide, the conditions of the tests being identical with those of the previous tests, except for the dissolved gas. The average temperature of water for all tests was 60 degrees F., with a maximum difference at any time of 4 degrees F. An analysis of water for CO_2 content was made before and after each test. Gas was added as necessary to keep the gas content approximately constant.

Of the five propellers tested, the results of tests from only one showed any differences that might be attributed to the presence of the dissolved gas. The results for the four remaining propellers showed no divergence other than might be due to experimental error.

Although the results of the tests with four of the propellers were consistent in showing no effect of the gas, the results obtained with the fifth propeller raised the question as to whether or not the interval of time in which a given particle of water saturated with gas moves in the low-pressure area is the factor which determines the effect that gas in solution may have on cavitation. This particular propeller, because of its characteristics, cavitated normally at a speed of advance approximately 70 per cent of the speed at which the remaining four propellers cavitated. Furthermore, the corresponding tip speed of this fifth propeller was about 80 per cent of that of the other propellers. Since the length of time that a given particle of water was in the low-pressure areas in the back of the blade is determined by these speeds, it follows that in the case of the fifth propeller greater time was available for the separation of the dissolved gas than was available with the other four propellers, and a greater liberation of gas might be expected. Further, cavitation in this instance was more general in extent. The greater spread of cavitation across the back of the fifth propeller perhaps meant greater permanency of the cavity formed, and with gases remaining longer in the cavity their effect may have been cumulative.

For the full-scale propeller the speed of advance and tip speed are increased above those of the model in the ratio of $\lambda^{\frac{1}{2}}$, where λ is the linear ratio between the model and full-scale propeller. The dimensions of the full-scale propeller are λ times greater than those of the model, likewise the length of the low-pressure region. Therefore at corresponding speeds the length of time a particle of water is in the low-pressure region on the full-scale propeller is $\lambda^{\frac{1}{2}}$ times greater than on the model propeller. Therefore it is possible that gas dissolved in water may affect cavitation in full-scale propellers and yet show no effect in tests with the model propellers.

COMPARISON OF RESULTS AND DISCUSSION.

From the results of our experiments thus far, it is apparent that back cavitation may first set in at either the tip sections or at any of the thicker blade sections. In propeller design it is therefore necessary that some relation between the cavitation index and the lift coefficient be established whereby cavitation can be avoided. To that end, experimental data have been plotted (Figure 1) in terms of a cavitation index and lift coefficient for the sections of the test propellers at which leading edge cavitation or "burbling" was first observed. In the case of leading edge cavitation the lift coefficients were calculated for blade sections at 0.9 of the radius from the center of the propeller. These data are compared with a theoretical curve derived by Lerbs (6).

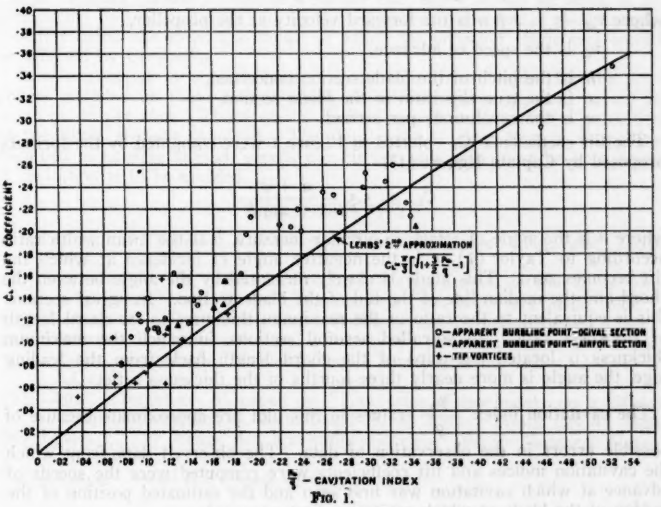

C_L — LIFT COEFFICIENT
$\frac{p_s}{q}$ — CAVITATION INDEX
LERBS' 2ND APPROXIMATION
$C_L = \frac{\pi}{2}\left[\lambda_0 + \frac{1}{2}\frac{p_s}{q} - 1\right]$
O — APPARENT BURBLING POINT—OGIVAL SECTION
Δ — APPARENT BURBLING POINT—AIRFOIL SECTION
+ — TIP VORTICES

FIG. 1.

In each case the cavitation index $\frac{p_s}{q}$ was determined as follows. p_s is the absolute constant pressure value at the shaft level that existed in the tunnel during test. This value is the sum of the atmospheric pressure (33

feet of water), the head of water above the center of the propeller shaft minus vapor pressure corresponding to the water temperature reduced to scale as required.

q is the velocity pressure, and was computed by $q = \frac{\rho}{2} v_r^2$, where ρ is the density of water, v_r is the relative velocity at the section of the blade at which cavitation was first observed, and is determined according to Eggert's method (1).

$v_r^2 = K\ (\pi\ f\ d\ n)^2$, in which $\pi\ f\ d\ n$ is the speed of the blade section expressed in feet per second. K is the secant squared of the angle between the direction of flow v_r and the plane of the propeller section.

In this, the angle of attack has been assumed to be that corresponding to half the slip, since the inflow velocity increment is assumed to be half of the velocity increase due to slip. Any error in this assumption will be of little consequence in the value of the relative velocity v_r, since the increment of the forward velocity due to the term $\frac{1}{2}\ s\ p\ n$ is a small fraction of the whole. f is the estimated fractional part of the propeller radius, at which "burbling" first became apparent. In the case of leading edge cavitation $f = 0.9$.

Then

$$v_r = [(\pi\ f\ d\ n)^2 + (v_a + \tfrac{1}{2}\ s\ h\ n)^2]^{\frac{1}{2}}$$

where $v_a + \frac{1}{2}\ s\ p\ n$ is the forward velocity at the propeller,

v_a is the speed of advance,

h is the pitch of the blade section concerned,
s is the true slip ratio of the blade section,
n is the revolutions per second.

The lift coefficients C_L plotted in Figure 1 were computed by the formula proposed by Captain Eggert (1).

$$C_L = 5.5\ \frac{a + c}{1 + 4.4\ b},$$

where a is the angle of attack in circular measure, b is the mean width ratio according to Taylor (7), c is the negative angle of incidence at which the lift becomes zero. This angle is closely measured by the angle between the chord and the median line of the tail of the blade section. In ogival sections this is equivalent to the ratio of the maximum thickness to its chord length or thickness ratio. In so-called aerofoil sections, in which the maximum thickness is located one-third of the chord length back from the leading edge, the angle is more nearly three-fourths of the thickness ratio.

The cavitation index $\frac{p_a}{q}$ values in this plot are approximate because of possible errors in the observation of data. The observed data from which the cavitation indices and lift coefficients were computed were the speeds of advance at which cavitation was first seen and the estimated position of the section of the blade at which cavitation first appeared.

In the same plot is given the equation derived by Lerbs (6) as being an approximation of the limit of lift relative to the cavitation index above which cavitation cannot be avoided. His equation is

$$C_L = \pi/3\ \left[\sqrt{1 + 3/2\ \frac{p}{q}} - 1\right]$$

In his paper (6) the author uses the "velocity of undisturbed flow" in computing the velocity pressure q. I understand this to be the speed of advance. Inasmuch as his approximate value of C_L is only a function of the cavitation index $\frac{p}{q}$, it is possible, for the purposes of comparison with our observed data, to plot his equation in terms of relative velocities, which has been done.

The apparent cavitation point of most of the model propellers plots above this approximate theoretical maximum lift. This may be attributed to the approximations, referred to above, in estimating the speeds and positions of sections at which cavitation first appears. It is quite probable that the cavitating point is reached at a speed at least one or two knots below that at which cavitation becomes apparent with sufficient definiteness to warrant recording. A corresponding increase in the cavitation index would cause the observed points to fall much closer to Lerbs' theoretical line. It will be noted that data obtained from propellers affected by leading edge cavitation plot nearer to the theoretical line than the data obtained by instances of "burbling." It is believed that this is due to the closer approximation of the correct speed at which leading edge cavitation began, and that no error was made in estimating the location of the blade section, at which cavitation developed. However, these differences as plotted are considered to be of little practical importance. From Figure 1 it is apparent that Lerbs' equation can be accepted for the present as a danger line, below which we shall, in all probability, be free of cavitation. It does, however, represent a good margin of safety below the speed and thrust at which the propeller efficiencies would be affected by cavitation.

It is interesting to observe here that the unit thrust values for the propellers included in these data at which cavitation became apparent varied as much as 50 per cent. There was a similar spread in the relative velocities at the blade sections involved. The data make it clear that in propellers designed to operate at high speeds the lift per unit area must be kept down if cavitation is to be avoided.

BIBLIOGRAPHY.

(1) Eggert. "Propeller Cavitation," Translation Soc. N.A. and M.E., 1932.
(2) Saunders. "The Variable Pressure Water Tunnel of the U. S. Experimental Model Basin," Transactions Soc. N.A. and M.E., 1930.
(3) Walchner. Hydromechanische Probleme des Schiffsantriebs. 1932.
(4) Ackeret. Experimentelle und Theoretische Untersuchungen über Kavitation, Technische Mechanik und Thermodynamic, Bd. 1, Nr. 1, S. 14, etc.
(5) Cook. Discussion of paper on "Cavitation Experiments on a Model Propeller," by Dr. G. Kempf and H. H. Lerbs, Trans. I.N.A., 1932.
(6) Lerbs. "Zur Frage des Kavitationseintritts," Werft Reederei Hafen, July, 1931.
(7) Taylor. Speed and Power of Ships. 1933.

DISCUSSION.

Dr. J. C. Hunsaker: It should be of interest to add something from the academic and scientific side to the insight on cavitation that our experiments give. At the Massachusetts Institute of Technology we have been con-

cerned for the last two years with research to determine the nature of cavitation, what makes it, and how it can produce its effects. To do this on turbine runners, or propellers, seems out of the question, because the phenomena are there too complicated. We have developed a technique borrowed from aeronautics to reduce the problem to a very elementary form. Through a Venturi nozzle in a pipe, water is forced at varying velocity, varying temperature, and varying air content. There are means for observing at leisure the distribution of pressure along a profile and glass sides to the experimental chamber or Venturi for viewing and photographing the phenomena.

We have found that the distribution of pressure along the profile is significant for cavitation. The average pressure of the stream falls, as it should, towards the throat to a low value, following Bernoulli's equation, and in the expanding region below the throat it will rise again almost to the original value, provided that there is no cavitation and continuity in the flow is preserved. If the flow be speeded up, or the static pressure on the whole stream be lowered until the mean pressure at the throat comes down to 20 mm. of water, the vapor pressure for room temperatures, the picture instantly changes, and instead of the pressure rising in the expanding portion it fails to rise, but remains low for some distance below the throat. At this same time we see a cloud of white steam forming, indicating that we have lost continuity. We have a flash from the liquid phase into the vapor phase in the region of low pressure, and, as the condition is aggravated by higher speed, this cavity of steam will extend farther downstream below the throat and, simultaneously, the low pressure will extend farther downstream. We have excellent correlation between the point of maximum rate of rise of pressure downstream and the visual end of the cavitation volume. The end of the cavitation volume is also opposite the point on the side walls where maximum damage is done to the metal when cavitation is present. The explanation seems to be that water vapor, formed under conditions of low pressure and room temperature at the throat, has proceeded downstream to a place where pressure is higher. It has gone against a rising pressure gradient and is virtually super-cooled. It is unstable and must go back into the liquid state, which it does at a particular region with considerable violence.

We have measured the sound and shock when cavitation is well developed and have a record of acoustic vibration, with heavy peaks of amplitude with a definite frequency of the order of 100 cycles per second. We photographed this white cloud of steam with Edgerton's technique of high-speed motion photography at five thousand pictures per second, and saw that the white cloud was not purely a white cloud, but vanished and reappeared with the same frequency as the shock measured by acoustic and pressure means. The volume of steam collapses back to water and gives rise to a series of trains of pressure waves which appear to cut off momentarily the generation of steam at the throat. The average pressure at the throat is the steam pressure, but one hundred times a second that pressure is higher and lower. The steam appears to be pinched off and turned on again, and so we have a trigger action with a consistent periodicity. Experiments with several profiles show this frequency to vary directly with the velocity and inversely with a linear dimension. The non-dimensional number $f\ L/V$ is substantially constant.

Cavitation effects we find always downstream and always at a preferred location. A profile will show cavitation, but do no damage if the collapse takes place in clear water and not in contact with the walls.

I suggest, in connection with marine propellers, that the observed loss of efficiency which comes with well-developed cavitation must be due to a change in pressure distribution such as we observed.

With higher air content in the water the violence of the cavitation damage is reduced. We know that with chemically pure water the collapse of vapor to the liquid phase is much more violent. We have made tests in which the damaging effect of cavitation is increased 400 per cent by reducing the air content of the water from 4 per cent to ½ per cent. Further experiments are necessary to establish the effect of air content quantitatively.

MR. G. S. BAKER, O.B.E. (Member of Council): There are several things in Commander Kell's paper which are of great interest to us. First of all, we are very much indebted to him for his descriptive notes with regard to cavitation as they have seen it in America; they are most informative and helpful. Secondly, we notice the support which the United States Government is giving in the form of a destroyer which he can use to carry on his work.

The author mentions that the tunnel which they are using is of the "open jet" type, so that you can see what is happening inside the tunnel. In case that description should lead anyone to think that with a "closed" jet you cannot see what is happening, I should like to say that you can see quite well, as we shall be able to show you on Thursday when you go to the Tank at Teddington.

I think that the most interesting part of the paper is the data given in Figure 1. Although I understand from Commander Kell that this figure is more or less of a general character, I would like to suggest that he should give the camber ratios of the sections which he has used in that work, as the diagram would then be of much greater value.

I should also like to ask him why he uses the calculated C_L from Eggert's not too well supported formula, as given in the paper, instead of actual values for the section? I know that we have practically no data for such sections in water, although we want them very badly, but I think that for the time being aeronautical data could be used, and it may be that Figure 1 would have a little different appearance if this were done. As it stands, it suggests that a circular-back section will take a higher C_L than an aerofoil section before it breaks down, and that is somewhat against our own experience.

One other matter is worth mentioning in connection with such figures. If v_m is the actual velocity of the water at that part of the back of a blade where burbling commences, this blade moving at a speed v_r, the condition for cavitation is that

$$p_a = \frac{v_m^2 - v_a^2}{2\,P}$$

and this gives

$$\frac{p_a}{q} = (a^2 - 1)$$

where a is the ratio v_m/v_a for this section at the working slip angle.

Lerbs' curve is based on the assumption that this ratio is sensibly the same as that over a circular-back section giving the same C_L—which can be easily calculated. But obviously this is only an approximation, since v_m/v_a varies with blade section, particularly at small slip angles. In fact, any such method of representation which ignores the fact that the base of Figure 1 is actually connected with blade shape is bound to show greater inaccuracy the smaller C_L becomes, as it is at small C_L values that the shape of section is almost predominant, and this is the main reason for the relative spreading of the spots in the figure at this end of the diagram.

HERR H. LERBS: We should be grateful to Lieutenant-Commander Kell for the complete results on cavitation tests which he has given us from the Washington Tank, as in some cases we have obtained rather different results in our Hamburg cavitation experiments.

On the question of face cavitation, our measurements have shown that face cavitation alone, without back cavitation, if sufficiently increased, leads to a diminution of efficiency, which also agrees with the principle of energy, because the bubble collapse is always followed by a loss of mechanical energy. In special cases of incipient back cavitation this loss of energy can be overcome equally well by increasing the lift coefficient caused by increasing camber ratio as by decreasing drag coefficient due to decreasing skin friction.

Very interesting are the systematic measurements of the influence of dissolved gases in the water upon the screw forces. We have not yet made in our Hamburg Tank any such systematic measurements; we have only made experiments with different screws in fresh tap-water, which at Hamburg is very full of air, and now we have measured thrust depending on times at constant values of vacuum and number of revolutions. We have always found that thrust increased with increasing time, till after some time a constant value was attained which could be always reproduced. The time depends naturally on the vacuum.

DR. D. P. RIABOUCHINSKY: In connection with the interesting communication of Commander C. O. Kell, I would like to mention some researches on the formation of cavities in liquids carried on at the Laboratory for the Mechanics of Fluids at the University of Paris.

These researches do not directly concern screw propellers, but the general theory which I have developed might also be applied to the latter.

We carefully examined first of all two specially simple cases, the strictly hydrodynamical solution of which could now be developed up to numerical values and be afterwards submitted to experimental verification.

The first case is that of an infinitely long circular cylinder suddenly set in motion perpendicularly to its length in a liquid initially at rest.

The second case is that of the same cylinder moving at a constant speed perpendicularly to its length at the moment when the pressure exerted upon the free surface, bounding the liquid externally, falls suddenly to a new and sufficiently low constant level.

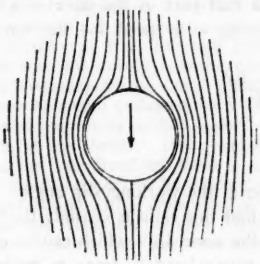


FIGURE 1.

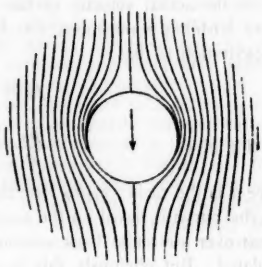


FIGURE 2.

Streamlines in the initial relative motion and the nascent cavities for both these cases are shown on Figures 1 and 2. The arrows show the direction of motion of the cylinders. The distribution of the initial pressure has also been calculated. (See para. (8) in Bibliography.*)

The general theory allows us to foretell that in the case of a sphere similar cavities must arise. In the first case the fluid will separate itself from the sphere on a surface surrounding the upper pole, and in the second case on an annular surface along the equator. The experiments which we have carried out have confirmed this theory.

As far as one can see from the above figures, the cavities arising would have to be washed away by the relative stream, and therefore the motion cannot tend to become permanent, but, as I mentioned in a previous paper (See (1), para. 28), it would tend, in Case 2, to become periodical.

The origin of periodicity in the experiments mentioned by Dr. J. C. Hunsaker could probably be interpreted as above.

The CHAIRMAN (Sir William J. Berry, K.C.B., Vice-President): I beg to move a hearty vote of thanks to Commander Kell for his valuable paper.

WRITTEN CONTRIBUTIONS TO THE DISCUSSION.

MR. J. F. C. CONN, B.Sc. (Associate-Member): Commander Kell has given an interesting paper on a difficult subject. It is regrettable that he confined himself to a discussion of cavitation in general terms, since it is scarcely possible to study the bahavior of any propeller without complete information regarding its geometrical properties, together with its thrust and torque performance over a range of slip.

His remark that "it is the back of the propeller blade and not the face that governs the propeller performance" can only be greeted with a certain mild surprise. The author does well to stress the possibilities of the time factor in model experiments as this is a point which may easily be overlooked.

We should be grateful to the author for any information regarding tests made on propellers with crescent blade sections, i.e., sections formed by the intersecting arcs of excentric circles. Aerofoils of crescent shape appear to have certain theoretical advantages, considered as blade sections for fast-running highly loaded propellers.

LIEUTENANT-COMMANDER C. O. KELL (CC), U.S.N.: I wish to thank the several gentlemen who have contributed to the discussion on this paper.

It is to be regretted that it was not possible for Dr. J. C. Hunsaker to project his slides and film which so clearly describe the work being done on the subject in the laboratories of the Massachusetts Institute of Technology. It was my privilege to see them last week at Cambridge, and they are most instructive. They show clearly the periodic nature of the phenomena.

Mr. Baker points out that our data plotted in Figure 1 are not complete in that we have not taken into account the blade shape in determining the abscissae of our points, in particular for those points of low C_L. I agree with Mr. Baker, and his point is important. However, at the present time we are not prepared to differentiate for the many possible variations in characteristics of blade shape. Further, I believe that the criteria upon which these data were based were sufficiently well defined to make it possible to show the effects of major changes in blade shapes.

I am very much interested in what Dr. Lerbs had to say relative to face cavitation. After further discussion with Dr. Lerbs, I do not believe the Hamburg and Washington tunnels are in disagreement on this point. Dr. Lerbs has measured loss in efficiency due to face cavitation at very low cavitation numbers, t, where $t = \frac{p_a - e}{q}$, and at very small positive and even

negative slips. Under such abnormal conditions of operation, face cavitation will be present to a much greater degree than can be expected in any reasonable design. I think Dr. Lerbs will agree that with conditions in his water tunnel which represent reasonable operating conditions on full scale, where face cavitation has been present, no effect could be determined.

The brief description by Dr. Riabouchinsky of his experiments at the Paris University is most interesting. His description of the formation of cavities on an annular ring along the equator of the cylinder in his second case checks our observations that the burbling type of cavitation has its origin at the position of maximum blade thickness. In this position, and at the equator of the cylinder in the second of Dr. Riabouchinsky's experiments, the relative velocities of water are maxima.

In reply to Mr. Conn, I regret that I am not at this time prepared with detailed information on the performance of propellers with crescent blade sections, together with the geometric characteristics of such sections. It will be noted from Figure 1 that the sections of most of the propellers tested thus far had the ordinary ogival back. There were a few propellers tested having the so-called aerofoil-shaped sections. It will be noted that as a rule propellers of this type cavitated at lower lift coefficients than comparable propellers with ogival-shaped sections. However, I do not feel that the relative merits in efficiency and non-cavitating characteristics of propellers designed with aerofoil sections and those with ogival-shaped sections have been sufficiently established at present to justify a statement.

* BIBLIOGRAPHY.

(1) D. Riabouchinsky: "Aperçu théorique sur la mécanique des fluides," Bulletin technique du Service technique de l'Aéronautique, No. 17, 1924.

(2) "Remarques sur le problème des cavitations," Comptes Rendus, t. 182, 1926.

(3) "Sur quelques cas de cavitations," Comptes Rendus, t. 184, 1927.

(4) "Théorie cavitationnelle de la résistance des fluides," Congrès International de Mécanique appliquée, Stockholm, 1930.

(5) "Sur quelques problèmes de mécanique des fluides relatifs aux parachûtes," I-er Congrès International de la Sécurité aérienne, Paris, 1930.

(6) "Recherches expérimentales sur la naissance des cavitations," Comptes Rendus, t. 195, 1932.

(7) B. Demtchenko: "Sur quelques applications bi-dimensionnelles de la théorie cavitationnelle de M. Riabouchinsky," Comptes Rendus, t. 192, 1931.

(8) Problèmes mixtes harmoniques en hydrodynamique des fluides parfaits, Paris, 1933.

(9) P. Léglise. "La technique et la recherche scientifique au XIII* Salon de l'Aéronautique," L'Aéronautique, Mars, 1933.

SAVING LIVES AT SEA.

By KENNETH PORTER, MEMBER.

I have read with a great deal of interest the article "Can We Eliminate the Fire Hazard?" reprinted from the "Marine Journal," of September 15, 1934, under the heading "Notes," appearing in the November, 1934, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. In this connection may I add a few thoughts that may possibly aid in overcoming the "hazard of the sea."

Accidents, whether preventable or unforeseen, will always happen, but the gravest danger of all is panic. Many lives have been saved by fireproof

construction, water-tight integrity, non-sinkable life boats and modern launching equipment, etc., but many more could have been saved if pandemonium had not broken out, people shrieking, diving overboard without a thought of how they may be saved. Life boats upsetting and dumping all their occupants into the sea; crashing against the side of the ship, and people racing to and fro, not knowing where to go or what to do—smoke filling their lungs and the hot deck scorching their feet. They surge to one side, causing the ship to list, then in their fright rushing to the other side, making it almost impossible to launch the life boats.

A large percentage, if not all, of this may be eliminated, by the use of a loud speaker system. In case of a fire; or hurricane in which part of the superstructure, some of the life boats and possibly the mast has been carried away with a tremendous crash; or in the case of an impending collision, is it any wonder that the passengers get hysterical? Just imagine what effect it will have on them to hear in a cool, calm voice, but with authority, "Your attention, please! There is a fire in Salon 'A,' but there is no danger. The bulkheads are fireproof. There will be a good deal of smoke, due to smothering of the flames, but do not get excited. Please go quietly to your cabins and put on the life jackets which you will find under your berth, then go up to the second deck and aft to the upper deck to the life boats assigned to you. Do not try to take your baggage; it will be safer in your rooms. The ship can't sink and there is no hurry!" The life jackets in addition to being buoyant should have a flame proof cover.

The old Viking ships and the old sailing ships weathered many a trying situation because their crews knew no fear. There are many people today who are "nearly scared to death" from the moment their feet touch the deck of a ship until they are on dry land again. They have been taught from childhood, by "tales of the sea," that the sea is a "cruel mistress." They are afraid that they might get sick. They are afraid of the rocking and pitching of the boat. They are afraid that it might sink.

Anything that tends to eliminate these fears and maintain order aboard ship in an emergency will do as much as any one thing to safeguard life. There are two things that will go a long way towards eliminating this fear and its accompanying panic.

First, by laws; second, by educating the passengers. A law should be passed requiring every ship carrying or having accommodations for more than twenty passengers and under way for more than twenty-four hours; and all ships carrying or having accommodations for one hundred people, regardless of the length of the voyage, be required to have a standard loud speaking system* with reproducers in the living quarters, dining rooms, salons and engine rooms for both passengers and crew. It's the unknown that frightens people the most. If any easy means is provided to acquaint them with the danger and at the same time instruct them what to do, most of their fears are eliminated.

The present regulations, regarding life boat drills at sea should be revised. Every ship leaving a port for more than twelve hours should be required to hold a life-boat drill within two hours after leaving port. The boats should be manned, uncovered and lowered to the rail. Every ship in the Navy has fire, collision, and abandon ship drills once a week, whether in port or at sea, with practically the same personnel aboard. Why should not the passenger liners be required to train their crews and safeguard the lives of their passengers in an emergency? It is true similar regulations regarding life-boat drills are now in effect but they are not enforced. I

* The loud speaker system was suggested by Lieut. Comdr. R. S. Barrett, De-F, U. S. N. R., during a discussion we had after the *Morro Castle* disaster.

have traveled on many passenger boats flying the American flag in which I neither knew to what life boat I was assigned, where it was located, or how to get to it.

There are three logical, if not safe reasons, why they are not enforced: (1) the penalty, unless there is an emergency is not sufficient; (2) often there is an insufficient number of the crew off duty to handle the life boats for drill purposes; and (3) the owners and the captain are afraid that the passengers will become unduly alarmed. They prefer not to cause them to even think of such a thing as the necessity of getting into a life boat.

The laws should be rewritten in a simple, clear, and understandable language that it will be impossible to evade. The penalty for failure to have the required drills should be the unconditional revocation of the captain's license and a severe penalty for the ship owners. Producing standard announcing systems in larger quantities will greatly reduce the present cost and will not work a hardship on the owners.

The next step is to educate the public that safety drills are required by law for their protection on every ship that sails the seas. If they are told when they buy their tickets at what time these drills will be held they will soon take it as a matter of course and realize that they are not just being inconvenienced. The boats, their rigging and equipment will be kept in first-class condition because the owners and the captain will know that it will be inspected by the passengers. Thus can a large measure of the panic, in an emergency at sea be eliminated and many lives saved.

PREVENTABLE FIRE AT SEA.

Many of the finest ocean liners afloat are firetraps! The personnel of many of them is inadequately trained and disciplined! In spite of these facts, fireproofing materials, fireproof construction, and practically 100 per cent protection against fires at sea are possible. Moreover, for American ships at least, there are available men of high character capable of maintaining not only our prestige on the seas but also the discipline and efficiency so necessary to the proper protection of every passenger aboard ship. This is the contention of R. G. Skerrett, noted writer on marine topics, who expresses much concern in an article in the current issue of "Scientific American," as to the way in which shipping executives have failed to take advantage of safety measures that are available.

"Will the horrifying loss of the S. S. *Morro Castle*," asks Mr. Skerrett, "lead to corrective measures that will make for greater safety at sea?" To answer this question he analyzes the situation by pointing out that both naval architects and shipping lines have been extremely shortsighted. To quote further: "The fire hazard in modern merchant ships, and especially aboard the de luxe liner, has led to extensive investigation, to the devising of numerous protective features and intrumentalities, and to a somewhat voluminous literature that is revealing of how much study has been given to the problem. Furthermore, various branches of our engineering and industrial life have contributed dependable apparatus for the early detection of fire, and have provided effective means for smothering or extinguishing flames, as well as structural materials that can be relied upon to resist the spread of fire long enough to give an alert personnel a very fair chance to bring the fire-fighting equipment into action effectually and decisively * * * a passenger ship can be made so far fire-resistant that flames can be held in check and the vessel safeguarded from extensive damage." Mr. Skerrett points out, however, that this fire-preventing and fire-fighting equip-

ment has not been adopted to the extent it should have been. In an effort to win patronage, the public spaces of some de luxe liners of today have many of the characteristics of accommodations found in overly elaborate hotels. The architectural embellishments are usually fashioned of wood and highly inflammable; while the interior decorator provides appointments that are more fuel for the flames. This state of affairs is, of course, ridiculous when there are available plastic materials for making fireproof wall panels and partitions, attractive metal furniture to take the place of ornate wooden decorations, and methods of fireproofing chemically such things as draperies and upholstery. Even treated wood has not been used to any large extent.

Intra-mural channels between the metal hull and the interior panelling contain fuel and also tend to promote a fire, once started, because of the drafts which may quickly carry the flames to all parts of the ship through these channels. Ways and means must be provided in structural design to restrict or choke these drafts, and furring or other carpenter work should be done only with treated wood.

Mr. Skerrett strikes a keynote in stressing the importance of discipline. This implies, first of all, careful selection of personnel of a high character. In the last analysis responsibility rests squarely upon the government and its statutory agency, the United States Steamboat Inspection Service. "The heartening aspect of the subject," Mr. Skerrett says, "is that we are positively assured that ships can be built without inviting excessive cost, that will be far better safeguarded against the hazard of fire than many of the liners afloat and in active service today. * * * Nevertheless, a grave weakness remains in the human equation of the problem; and only an aroused public opinion and popular insistence will bring about the necessary corrective measures and their enforcement. * * * Our Merchant Marine, like the Navy should be able to live up to any prescribed standards."—"Scientific American," January 18, 1935.

SAFETY IN MARINE TRANSPORTATION.

The Bureau's activities having a bearing on safety requirements for ship construction, equipment, cargo, and stowage, and on aids to navigation, have included a considerable range of subjects, some of which are briefly indicated below. Further items thereon will be given in succeeding issues. The projects or tests were undertaken mainly at the request or with the cooperation of the Bureau of Navigation and Steamboat Inspection, the Bureau of Lighthouses, the United States Shipping Board, and groups of marine underwriters.

Fire resistance in ship construction.—The use of steel in hulls, decks, and bulkheads, while effecting a decided increase in safety compared with wood construction, would not as such be expected to prevent the occurrence of disastrous fires to any greater extent than incombustible walls, partitions, columns, and floors in buildings. In both cases the spread of fires in combustible furnishings, trim, and contents must be restricted by vertical and horizontal barriers. Many measures accepted as necessary in building construction have not been applied in the construction of ships, mainly on account of added weight and cost. Recent developments in light-weight incombustible or fire-retardant materials have overcome these objections to a considerable extent.

The construction of fire-resistive bulkheads transversely across the ship has received recognition in recent years as essential in restricting horizontal spread of fire. According to the regulations adopted by the International Convention for the Safety of Life at Sea, these are required to be spaced

at intervals not exceeding 40 meters (131 feet), and to be effective as a barrier for 1 hour against fires producing temperatures of 815 degrees C. (1500 degrees F.) at the bulkhead. Uninsulated steel bulkheads have been advocated for acceptance under this requirement, although obviously not meeting it because of ready transmission of heat through the metal. The results of fire tests as well as construction requirements have indicated, however, the desirability of incorporating the steel bulkhead as a part of the construction, with incombustible heat-insulating materials applied on both sides. For the 1-hour requirement, the total thickness of such materials need not exceed 2 inches and the weight need not be greater than 10 pounds per square foot. In building construction, fire barriers of comparable importance would be required to restrict the spread of severe fires for 4 hours. With proper choice of materials this can be effected with a thickness of 4 to 5 inches and weight of 20 to 30 pounds per square foot. As applied in ship construction the hull, decks, longitudinal bulkheads, and intersecting pipe lines would need insulation for a foot or two on each side of such bulkhead to prevent transmission of fire from heat conduction through the metal. Constructing intermediate longitudinal and traverse fire-resistive bulkheads and cabin partitions is considered desirable as an aid in further restricting the spread of fire. On account of limitations in thickness and weight, the fire resistance that can be practically attained for them would be in the range one-half to one-fourth hour as gauged by the standard fire-testing procedure for fire partitions. Even so, as built of incombustible or fire-retardant materials they would serve acceptably in confining the fire until detected and extinguished with the equipment provided.

Openings in the main fire-resistive bulkheads need protection with automatically closing or self-closing fire doors. Stairways, particularly those communicating between more than two decks, need fire-resistive enclosures with lighter self-closing fire doors in the openings. The number and location of these stairways should be such as to afford egress to relatively safe areas in case of fire. Fire stopping the spaces between the deck and bulkhead plates and their linings at suitable intervals will prevent rapid spread of fire through such concealed channels.

Because of weight limitations, deck construction cannot be given as high a degree of fire resistance as floors in fire-resistive buildings. Nevertheless, to judge from results of fire tests with unprotected and protected welded steel floors, protection comparable with that given by one-hour bulkheads is obtainable with incombustible or fire-retardant linings over the beam flanges. These should be one-half to one inch thick, combined with an incombustible top finish over the deck plate of comparable thickness. These linings and finishes would replace the combustible boards and floor covering otherwise used in spaces occupied by passengers and crew. Mastics with low combustible contents and excellent strength, bonding, and wearing properties are now obtainable at moderate cost. They can serve as the finished wearing surface or as a base for the same.

Fire-retardant treatments.—The general methods for treating wood to reduce its flammability involve impregnation with chemicals under pressure. No treatment has been developed that has the desired permanence as exposed to the weather, without protective coatings, but for interior use a number of them have been in practical use for the past 30 years or more. Specifications, inspection, and tests can be applied that will give fair assurance of obtaining a required fire-retardant effect. A number of other requirements must be met by the material, including freedom from unduly corrosive effects on metal, unduly poisonous chemicals, hygroscopic chemicals, and surface discoloration. A full treatment increases the weight of the wood about 15 per cent and the cost of hardwood finishing lumber about 100 per cent. Where

combustible materials are prohibited by building ordinances, the product has found considerable application for finish flooring and interior trim, although in recent years displaced to some extent by metal trim and composition floor finishes.

Surface treatments of wood have only minor value in decreasing the flammability, on account of the thinness of the protecting coating. Ordinary oil paints with a relatively high mineral pigment content have been found as effective as so-called "fire-retardant" paints.

Draperies and curtains can be satisfactorily flame-proofed by spraying or dipping, using a number of chemical mixtures. A borax-boric acid treatment developed for treating theater scenery has been found very effective, and the cost of the chemicals is relatively small. Such treatments must not deteriorate the fabric nor cause change in color or luster. The ordinary treatments are not permanent under laundering, although a treatment has been developed for cotton goods that gives fair resistance from this standpoint.

Satisfactory heat-insulating and sound-absorbing media can now be obtained of incombustible materials, and some of the combustible materials can be treated with admixtures, surface coatings, or veneers to reduce the flammability.

Cargo and stores.—The properties considered hazardous include low flash, fire, and boiling points for liquids, low ignition temperature, high flammability, and susceptibility to spontaneous heating, liberation of oxygen, and detonation. Tests for such properties have been made of a wide range of materials submitted for ship cargoes and stores, necessitating in some cases extended experimental inquiry, among which may be mentioned the work on the effect of heat on celluloid (Bureau of Standards Technologic Paper No. 98, now out of print, but available at many technical libraries) as well as the heating and ignition tests with jute (Monograph, National Fire Protection Association, 1934).

Fire detection.—According to an Act of Congress, all vessels more than 150 feet in length contracted for after June 30, 1916, which are provided with staterooms or other sleeping quarters for passengers, are required to be equipped with automatic fire-detecting systems for all spaces not readily open for inspection. In case of fire these detecting devices must give audible and visible alarm signals at the bridge and other points where a continuous watch is maintained. This constituted a considerable advance in safety requirements over those of many other maritime nations. Under this requirement tests have been made of the thermostats and other thermosensitive elements employed. The general designs of various systems have also been studied. As a result of these tests decided improvements have been made in sensitivity and reliability of operation of these devices, so that it is now possible to receive positive warnings of a fire at an earlier stage than formerly, and there is less likelihood of failure.

Motion-picture projection.—The showing of nitrocellulose motion-picture film was formerly prohibited on ships of United States registry. The small range in subjects obtainable with the slow-burning film and the difficulty experienced by inspectors in distinguishing between the two types led to the adoption of safety measures that enabled the flammable film to be shown. Recommendations were drawn up for properly designed and vented projection booths, and examinations made of motion-picture projectors for needed safety features. While it is more difficult to obtain a desired degree of safety on shipboard than in theaters on account of the use of portable booths and screens, the records show no instance of fire or jeopardy to life resulting from the showing of motion pictures during the 12 years that these regulations have been in effect.—U. S. Department of Commerce, "Technical News Bulletin of the National Bureau of Standards," December, 1934.

COMPARATIVE COST OF MOTOR SHIPS AND STEAMSHIPS.

"When the Holland-Amerika Company decided recently to construct two new ships for mixed passenger and freight service, between Holland, England, and South Africa, a serious study was made with a view to determine the expenses of building and operating ships having the following three types of machinery: 1—Diesel engines; 2—high pressure turbines; 3—low pressure turbines.

The contract of construction called for the ships to maintain a regular service in the following rotations:

24 ½ days at sea at a speed of 16 knots.
21 ½ days at a speed of 14 knots.
28 ½ days in port.

The first studies showed that as a consequence of the smaller space occupied by the motor and the supply of fuel oil, the Diesel engine ship could be slightly smaller than the steamship. Under these conditions the cost of construction of the motor ship was $10,500 less than the high pressure turbine ship and $30,500 less than the low pressure turbine ship. Allowing a percentage of 15 per cent a year for amortization, interest, insurance, etc., the annual running expenses for the motor ship were $2,000 less than for the low pressure turbine ship and $4050 less than the high pressure turbine ship. The amount of fuel oil expended for the complete trip was the following:

Low pressure turbine	3000 tons fuel oil
High pressure turbine	2500 tons fuel oil
Motor ship	1350 tons fuel oil

The expense for the above fuel oil was:

Low pressure turbine	$11500
High pressure turbine	$ 9600
Diesel motor	$ 7200

In résumé, the annual cost of fuel oil for the Diesel motor was $24,000 less than the high pressure turbine and $43,000 less than the low pressure turbine.

On the other hand, as regards the expense of personnel the motor ship was more costly, annually, to the extent of $3400.

Finally, the Diesel motor ship was adopted because it permitted a reduction of annual charges of $22,500 as compared to the high pressure turbine ship, and $44,000 as compared to the low pressure turbine ship."—Translated from the "Journal de la Marine Marchande," December, 1934.

ELECTRIC PROPULSION SYSTEMS.*

A Survey of Turbo-electric and Diesel-electric Drives and Methods of Control, Their Applicability and Respective Operating Characteristics.

Present-day practice has resolved the variety of schemes which have been put forward for electric ship propulsion during the past 25 years into two classes, namely: (1) turbo-alternators with synchronous motors, for large ships; and (2) Diesel engines with D.C. generators and motors, for small

* Abstract of a paper entitled "Generation, Distribution and Use of Electricity on Board Ship," by C. Wallace Saunders, H. W. Wilson and R. G. Jakeman, read in London before the Institution of Electrical Engineers on November 22.

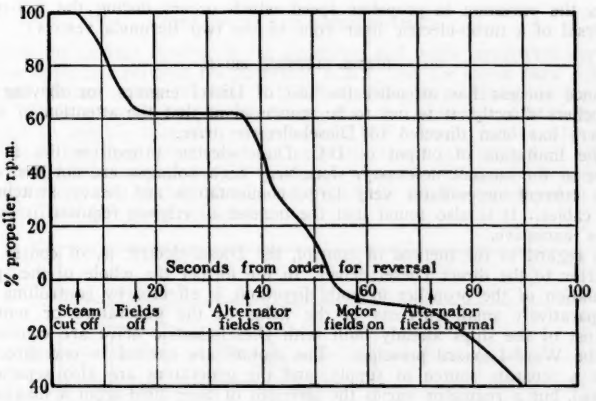


FIG. 1.—FULL-SPEED REVERSAL OF TURBO-ELECTRIC VESSEL.

ships. The line of demarcation between these two categories is not very clear. The maximum horsepower permissible with direct current is limited by the current to be carried by the commutators, control gear, and cables. On the other hand, at this limit, the maximum economical limit of the Diesel engines has not been reached. An attempt to bridge over the gap between the two methods is the suggestion to use Diesel engines with A.C. generators, and synchronous motors. This has not yet been put into practice, but may have possibilities in the future.

TURBO-ELECTRIC DRIVE.

For very large ships there is little doubt that the turbo-electric drive is the most suitable at present. With this scheme, the A.C. generators are direct-coupled to turbines and supply 3-phase current to synchronous propulsion motors.

The frequency at normal speed has been usually fixed at 50 cycles per second for ships built in this country, probably because it is the standardized frequency for land practice. Actually, there is no reason why the frequency should not be chosen at any value, having regard to the speed of the turbines and alternators and to the windings of the alternators and motors. In the French ship *Normandie*, the frequency will be 81 cycles per second, the alternators running at 2430 R.P.M.

The scheme of control for A.C. drive using synchronous motors is briefly as follows. Speed control is obtained by varying the frequency of supply, *i.e.*, by varying the speed of the turbine driving the generator. This can be readily carried out by means of levers or wheels on the control board through the medium of oil relay systems or servo-motors, which alter the governor setting of the turbine.

Since large starting and reversing torque is required for quickly accelerating and stopping the propellers and the vessel, the motors are usually started as salient-pole induction motors, afterwards synchronized at low speeds, and then adjusted to whatever running speed is required. The control gear is arranged at a central position in the engine room, and takes the form of handwheels or levers for each turbo-generator. Figure 1 indi-

cates the variation in propeller speed which occurs during the full-speed reversal of a turbo-electric liner (one of the two Bermuda vessels).

DIESEL-ELECTRIC DRIVE.

Since success has attended the use of Diesel engines for driving the propellers directly, it is not to be wondered at that the attention of shipowners has been directed to Diesel-electric drive.

The limitation of output of D.C. Diesel-electric propulsion lies in the value of the current necessary, since very high voltages are not desirable. This current necessitates very large commutators and heavy switchgear and cables. It is also found that the number of engines required is sometimes excessive.

In regard to the method of control, the Diesel-electric is, of course, far superior to the direct Diesel drive. In the former the whole of the speed regulation of the propeller in both directions is effected by controlling the comparatively small currents in the fields of the generators or motors.

Most of the ships already built with Diesel-electric drive are controlled on the Ward-Leonard principle. The motors are excited in one direction from a separate source of supply, and the generators are also separately excited, but a regulator varies the strength of their field from a maximum in one direction to a maximum in the other. For small twin-screw boats using two engines, it is sometimes necessary to run both motors from only one engine, in which case arrangements may be made to reverse the motor fields after reducing the generator voltage to zero. At starting a small excitation is applied to the generator, producing a small voltage which sends a current through the motor armature. Since the motor field is excited, a torque is exerted which turns the propeller. As the speed rises, the field of the generator is progressively strengthened until the desired speed is obtained. It will be seen that if the generator field is strengthened too quickly before the propeller speed has had time to rise, execessive currents may flow which may cause dangerous sparking at the commutators, and possibly overload the engines.

DIRECT AND FOLLOW-UP CONTROL.

The two principal methods of control on the Ward-Leonard system are known as the "direct" and "follow-up" methods. In the direct method, with a cam-operated controller or a face-plate controller, the regulator in the generator field is directly operated from the bridge by means of mechanical drive, the controller taking the form of an ordinary ship's telegraph. The regulator has a comparatively small number of sections (eight or ten) in the case of the former, but a much larger number in the case of the latter type. The follow-up method of control operates on the Wheatstone bridge principle. The telegraph on the navigating bridge moves an arm on a potentiometer regulator. When this arm is displaced from the position of the corresponding arm of another potentiometer mounted on the main regulator, a circuit is completed which starts a small motor driving the main regulator and keeps it running until the arm on the second potentiometer again corresponds to the arm on the bridge. The motor then stops. Thus the regulator in the generator field cannot possibly be operated too quickly, however rapidly the telegraph is moved.

In the Ward-Leonard system, if the generators are shunt-wound or lightly compound-wound, there is no danger of the propellers racing in a heavy sea. If the load comes off a motor, its speed rises a small percentage until its back E.M.F. is equal to the voltage of the generator and the current falls to practically zero.

OVER POWER AND REVERSE POWER.

Since the current flowing in the generator and motor armatures depends on the difference between the generator E.M.F. and the motor back E.M.F., a small change in one of these E.M.F.'s makes a large change in the current. It is therefore necessary to see that the generator E.M.F. is not altered so quickly that the motor back E.M.F. cannot follow. In small vessels it has been found that the propeller speed can change so rapidly that very little over power is produced when the telegraph is suddenly thrown from stop to full ahead, and very little reverse power when it is thrown from full ahead to full astern.

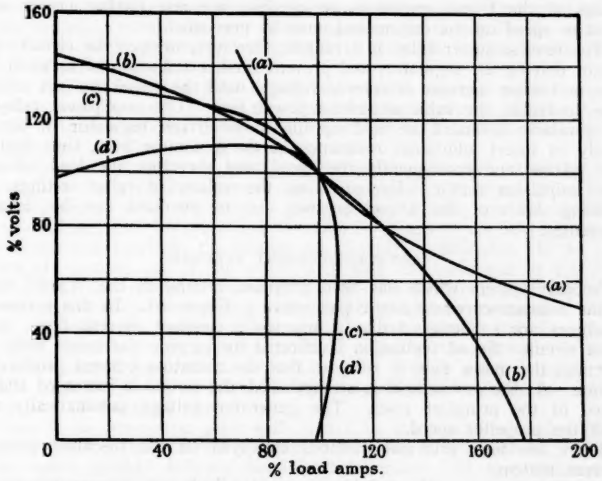


FIGURE 2.—GENERATOR CHARACTERISTICS FOR DIESEL-ELECTRIC PROPULSION.
(*a*) Theoretical characteristic for constant engine power.
(*b*) Constant-power system.
(*c*) Constant-current system.
(*d*) Austin system.

In order to avoid the necessity of providing the engines with a large margin of power, some form of over-power relay may be fitted which inserts a resistance in the generator fields if the power taken exceeds the normal power of the engines. In this way a heavy current may be taken at starting, when the voltage is low, but the current is limited when the voltage has risen.

Constant-current systems (described later) automatically take care of this question of overloading to a certain extent.

An alternative system which definitely prevents overloading during turning is called the constant-power system (curve *b*, Figure 2). This system consists of a separately-excited generator with a heavy counter-compound winding which has a drooping characteristic and can be so designed that the power supplied (volts × amperes) is approximately constant over a small range, and is equal to the engine power. Thus an overload due to

turning the ship is prevented by the fact that as the voltage falls the current rises. A disadvantage of this generator is that the open-circuit voltage is rather high, so that there is a tendency towards propeller racing in rough seas. This can be overcome by means of a voltage relay which inserts resistance in the generator field if the voltage exceeds a predetermined value.

For large vessels it may be found advisable to adopt the use of motor-driven field regulators, adjusted to operate over their whole range from full ahead to full astern as rapidly as possible, and to provide over-power and reverse-power relays (which can readily be combined in the same relay). Reverse power up to the value which will absorb the friction and driving losses of the Diesel engine is permissible, but any further power would tend to speed up the engine and must be prevented.

The reverse-power relay is arranged, therefore, to open the circuit of the motor driving the regulator, and prevent further reduction of forward voltage, or further increase of reverse voltage, until the vessel has lost sufficient way to reduce the value of reverse-power feed. The over-power relay can be similarly arranged to hold up the motor-driven regulator or alternatively to insert additional resistance in the generator field, thus reducing the voltage and consequently the speed, and therefore the load taken by the propulsion motor. By adjusting the values of relay settings, etc., slowing down of the Diesel engines due to overload can be definitely prevented.

CONSTANT-CURRENT SYSTEMS.

Another system which has been proposed instead of the Ward-Leonard is the constant-current system (see curve *c*, Figure 2). In this system the machines are so designed that a practically constant current flows in the main circuit. Speed regulation is effected by varying the motor field. At starting, the motor field is zero, so that the armature current produces no torque. As the motor field is strengthened, the torque is increased and the speed of the propeller rises. The generator voltage automatically rises with the propeller speed.

There are four principal methods employed on the so-called constant-current system.

(1) The first uses the 3-field generator, which has a separately-excited winding, a self-excited winding, and a counter-compound winding. This machine gives a practically constant current over the working range, but the open-circuit voltage is rather high.

The condition of the propeller coming out of the water is entirely different from that of the Ward-Leonard system. The generator is designed to give constant current, and therefore continues to supply this current. The motor therefore continues to exert torque with the propeller out of the water. Consequently, the speed rises until the counter E.M.F. of the motor is equal to the open-circuit generator voltage. It is a simple matter, however, to arrange a relay which will open the separately-excited generator field when the propeller speed rises a small percentage above the normal. This cuts down the current, and with it the torque.

With the 3-field generator it is possible to arrange the windings in such a way that the current carried when the voltage is reversed is considerably reduced. This condition occurs when the telegraph is moved quickly from full ahead to full astern. When the motor field is reversed with the motor still running ahead, the voltage is reversed and power is taken from the propellers and absorbed in driving the engines. Under this condition, the generator can be designed to take such a current that the reverse power is only that required to run the engines light at their normal speed.

(2) In the Austin system (curve *d*, Figure 2) the generator has specially-shaped pole shoes. The characteristic is similar to that of the 3-field generator over the working ranges, but the open-circuit voltage is lower. It is claimed that the speed rise with the propeller out of the water is not excessive.

(3) A third system employs standard generators and uses a current relay which automatically regulates the generator field. If the current tends to rise, the generator field is weakened, and *vice versa*. With this system it is necessary to limit the generator voltage in order to prevent racing.

(4) A fourth system has been called the controlled current system. In this system a 3-field exciter is used, the counter-compound winding of which carries the main motor current. The characteristic is exactly the same as that of the 3-field generator. Regulation is effected on the exciter field instead of on the generator field.

In a single-screw ship with a constant-current system, more than one generator can be used, as for the Ward-Leonard system. In the 3-field and the Austin methods, the generators are connected in parallel, and in the relay method they are in series.

In a twin-screw ship the port and starboard sides can either be kept separate, or the propeller motors can be connected in series. The latter method presents some advantages, and the two motors can be controlled entirely independently in speed and direction by means of the field regulators. In a rough sea, however, the method may involve a difficulty. If the propellers are coming out of the water they may not come out at the same time. When one propeller comes out, it is necessary to reduce the armature current and prevent racing. Since the two motor armatures are in series, however, the current in the other is also reduced and the driving power of the ship is lost. When the first propeller has entered the water again, the second may have come out, and the driving power is again lost. In this way, the power to drive the ship may only be available for a short proportion of the total time.

There is an interesting point with regard to the operation of the Ward-Leonard and constant-current systems. With the Ward-Leonard, the propeller speed quickly follows the telegraph orders. If the telegraph is changed from full ahead to half ahead, the regulator is quickly changed to its half ahead position. Reverse power will flow from the motors to the generators, and the propeller speed will be quickly reduced.

The same change of order with the constant-current system will quickly reduce the motor-field regulator to the half ahead position. The torque is therefore reduced, but it remains in the same direction. The propeller speed will fall to slightly above the trailing speed and will gradually fall to half speed as the ship loses way. This difference in response will, of course, be readily recognized by the navigator. If a quicker response is required with the constant-current system, it is only necessary to put the telegraph astern until the ship loses way, and then come back to half ahead.—"Shipbuilding and Shipping Record," December 6, 1934.

THE BIROTOR FIXED-VANE ROTARY PUMP.

The rotary pump, though an old mechanical device, is by no means obsolescent, and its field of usefulness is extended from time to time by new adaptations. The latest of these is illustrated in Figures 1 and 2, which show the "Birotor" pump manufactured by Messrs. Metropolitan-Vickers Electrical Company, Limited, Trafford Park, Manchester, under license from

Messrs. Société Anonyme Birotor. As the name implies, the pump has two rotors. One of these is inside the other, the inner rotor being driven from the outer one by means of a single vane attached to the outer rotor. We have used the term "fixed vane" in our title to emphasize the fact that the vane is not of the radially-sliding type commonly associated with vane rotary pumps. The pump functions by variation of the space between the outer and inner rotors. It is, therefore, of the positive displacement type and, in addition to sharing, with other forms of rotary pump, the advantage of absence of valves, it has a high mechanical and volumetric efficiency, which normally exceeds 70 per cent. It is particularly suitable for dealing with viscous fluids, and, apart from general service requirements, special attention has been paid to the design of the pump for service with fuel oil burners, an application in which uniformity of pressure is essential. Volatile fluids can be handled by the pump, as it is self-priming. A suction lift of 25 feet can be easily maintained when non-volatile fluids are being pumped.

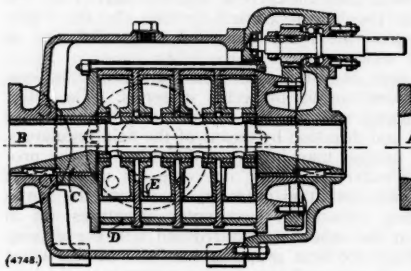


FIGURE 1.

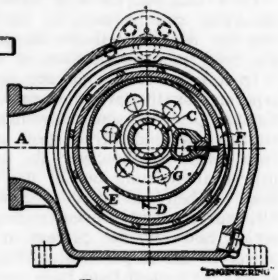


FIGURE 2.

THE BIROTOR FIXED-VANE ROTARY PUMP.
CONSTRUCTED BY MESSRS. METROPOLITAN-VICKERS ELECTRICAL COMPANY, LIMITED.

Dealing with the construction of the pump, it should first be realized that the rotors do not extend continuously from end to end of the chamber. They are divided longitudinally into "cells," which, however, are not connected in series, but work in parallel, the object being to ensure a steady discharge by arranging the ports of each cell in different positions on the periphery. It will be clear from the longitudinal section given in Figure 1 that the pump shown there has four cells. The suction branch is at A, the discharge taking place at branch B, that is, through the hollow shaft. It will be noticed that there is an opening at both ends of the shaft. A double discharge can be thus arranged for, but, more usually, the opening at the right hand is blanked by a cover and is used for inspection, etc. The shaft C is keyed to the casing and is in three parts for convenience of assembly, the center being keyed to the two end portions, and its axis being eccentric to that of the end parts. The outer rotors, D, are held between discs rotating on the outer ends of the shaft and connected by longitudinal bolts. The rotor cells are separated by partition discs clear of the shaft. The inner rotors, E, rotate on the central part of the shaft. The end discs and inner rotors are provided with bushes.

The outer rotors are driven by a spur ring attached to the right-hand disc and meshing with a pinion mounted on a shaft running in ball bearings and connected to a motor or other means of drive. It will be realized that the

whole of the rotating parts work in the fluid being pumped. The only gland required is on the suction side and is thus subject to a low pressure only. No separate internal lubrication is necessary even when petrol and similar liquids are being handled, a feature which eliminates risk of contamination of the fluid, such as food products, by lubricating oil. Referring to the cross section of Figure 2, it will be seen that the outer rotor is provided with a single port F, and the inner rotor with a single port G. The vane traverses the space between the rotors just below port F and just above port G. It is permanently attached to the outer rotor in a radial position and is also held in grooves in the partition discs and the end discs. Its function is two-fold. In the first place, it forms the driving member for the inner rotor in which, owing to the method of support, it is not subject to any appreciable bending stress. As the two rotors rotate round different axes there is a constant variation in the position of the point of entry of the vane in the inner rotor and in the angle of entry. This variation is met by a pair of semi-cylindrical bushes which allow both of sliding and angular movement. The second function of the vane is to form a division between the suction and discharge portions of the space between the rotors, the other point of separation being effected by the permanent rolling contact between the rotors on the top center line.

The whole space between the rotors is crescent-shaped in cross section. In the position of the rotors shown in Figure 2, the space is divided into a small suction space connected with the pump chamber by the port F, and a larger discharge space connected with the interior of the inner rotor by the port G. The interior of the rotor is in communication with the discharge branch through ports in the central part of the shaft. The direction of rotation is indicated in Figure 2. By imagining the vane to be travelling in this direction, it will be realized that the volume of the smaller suction space will steadily increase while the volume of the discharge space will as steadily decrease and the fluid contained in it will be forced through the discharge opening. By the time the vane has reached the top center, the ports in both the rotors have been closed as each is blanked by the wall of the other in the area of virtual contact at each side of the top center. Discharge has then finished and the crescent-shaped space is fully charged. On further rotation the discharge of the contents of this space begins and, the suction port opening, a new charge is taken into the developing "other half," as shown occurring in Figure 2, in which the vane has traversed 90 degrees from the opening point. If a single cell pump is employed, as it sometimes is, for light duties, the discharge is variable, but it is obvious that, by multiplying the number of cells and distributing the pairs of ports in the cells all round the full circle, a practically constant volume discharge is obtained. The maximum variation of torque during one complete revolution of a pump having two or more cells amounts to about 3 per cent only. The starting torque is of the order of 1.5 times the full load torque. The pump is made in various sizes from a single cell hand-operated pump to power-driven pumps delivering up to 50,000 gallons per hour, and discharging at pressures up to 300 pounds per square inch. These limits can, however, be exceeded when necessary, by modification in design.—"Engineering," December 7, 1934.

WEAR TESTING MACHINE.

In general it may be said that the resistance to wear of metallic articles does not depend only on the strength of the material nor only on its hardness. Hardness indicates durability against wear only if materials having the same composition and heat treatment are compared. It is, however, impossible

to judge—according to the value of hardness of substantially different materials—what relative ability to resist wear they have. In such cases the materials should be individually tested for resistance to wear. In the Skoda-Sawin machine, marketed in this country by the Selson Machine Tool Company, Ltd., of 23 Abbey House, Victoria Street, London, the wear is tested by means of a standard rotating disc of Widia. It offers the possibility of quickly testing the surface in less than five minutes and expressing the resistance against wear with a sufficient measure of probability in units of volume of the impression in thousandths of a cubic millimeter, which has been attained after a specified number of revolutions of the disc. The disc works at a specified load and circumferential speed. The impression caused by the disc does not, it is claimed, damage the surface of the tested article and the test itself can be carried out in the workshops in a similar way to that of carrying out tests of hardness on the Brinell, Rockwell, and Vickers apparatus.

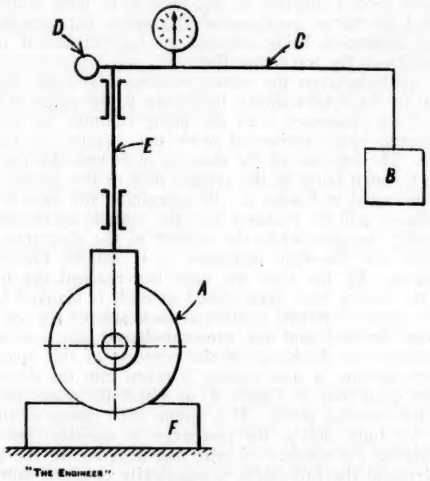


FIGURE 1.

The principle of the machine is shown in Figure 1; the disc A is supported on two bearings, connected by a yoke. The weight B acts on this yoke through a lever C, which swivels around fulcrum D and through a rod E, which slides in bushings. The disc is driven by a motor, located in the base of the machine. The power is transmitted from the motor by means of a belt pulley and a short flexible shaft. The test piece F is cooled by a stream of suitable liquid, which also carries away the chips formed under the disc. The test piece is clamped in a universal vice, which can move up and down in the guide of the stand of the machine. Before the test is started the vice is levelled in a direction at right angles to the axis of the disc by means of a spirit level, and by lighting up the gap between the test piece and the disc the surface of the test piece is brought accurately parallel with the periphery of the disc. When the piece has been pressed

up to the disc and the pointer of the depth gauge is in the zero position the motor is started; at the same time also the small pump for cooling liquid arranged in the base is put into operation. The penetration into the piece is observed by the depth gauge, and the revolutions of the disc are counted. After a specified number of revolutions the machine is stopped, the vice is let down and a microscope mounted in a joint at the left side of the frame is swung over the impression. A sliding shutter in the microscope makes it possible to measure the width of the impression in units of the microscope and the value of impression is determined from a table in thousandths of a cubic millimeter.

If pieces of hard surface are tested, such as cutting tools and measuring instruments, tools of Stellite and Widia, case hardened or nitrided machinery parts, the Widia disc should be 25-30 millimeters in diameter, 2½ millimeters wide, and should run at 675 R.P.M. and at a pressure of 15 kilos. For cooling, a ½ per cent solution of potassium dichromate in distilled water is used. The total number of revolutions of the disc working continuously should be 3000 ± 10, which corresponds to a duration of the test of about 4½ minutes.

In a comparison of the wear of Zeiss block gauges of tool steel and of Poldi maximum high-speed steel, tested by the Skoda-Sawin machine, it is interesting to find that the gauges of high-speed steel had an amount of wear equalling 40, whereas the gauges of carbon steel of the same Vickers hardness, 802, had a much greater amount of wear, *i.e.*, 106. This relation is borne out in practice.

As the volume of the impression increases proportionally with the number of revolutions of the disc, or with the period of duration of the test, it is possible to carry out by this machine tests on all kinds of material. If tests of soft materials are to be carried out, the duration of the test can be shortened and the results converted to the normal number of revolutions of 3000, which have been fixed for the tests of hard surface pieces. Tests can also be carried out on the basis of a constant volume of impression observed by a depth gauge, in which case the wear is measured by the period of duration of the test.

It has been ascertained that even the slightest change of composition of the cooling liquid has an influence upon the results and that the Skoda-Sawin machine can consequently be used with advantage also for testing various kinds of lubricating oils, emulsions, and solutions used as cutting liquids.—"The Engineer," December 7, 1934.

LUBRICATION.

EFFECTS OF SIDE LEAKAGE IN JOURNAL BEARINGS.

Theoretical treatment of journal bearing lubrication, based on an infinitely wide bearing (hence no oil flow from the film sides) has been highly developed. Less is known, however, of phenomena that occur in an actual oil film. Experiments necessary on a bearing of finite proportions and from which oil is leaking at the film sides show results entirely different from those predicted by theory which takes no account of side leakage.

By an indirect method of electrical integration, Sidney J. Needs undertook an analysis of the problem as applied to a 120-degree centrally supported bearing, the results of which were presented in "Effects of Side Leakage in 120-Degree Centrally Supported Journal Bearings." Effects of side leakage are found to vary not only with bearing length and width, but also with load. Results are given in tabular form and also by curves, and from these, best operating conditions are apparent.

JOURNAL BEARING TESTS.

Although the hydrodynamical theory of lubrication was developed long ago, design of modern bearings is still based largely on empirical data. Theoretical developments treat an ideal bearing never realized in practice. Even for this ideal bearing the calculations are complicated, particularly for journal bearings. Deformation of the bearing by loads and temperature stresses, accuracy of machining and degree of finish are factors which cannot be easily introduced in theoretical consideration. For these reasons, improvements in bearing designs can be made only after data are obtained from tests duplicating as much as possible normal operating and manufacturing conditions.

Tests made on journal bearings by Westinghouse were the subject of "Performance of Large Journal Bearings," by R. Baudry and L. M. Tichvinsky. These showed rather close agreement with theoretical results made available in recent years. Laboratory tests for determining maximum carrying capacity and coefficient of friction for various speeds were made using a standard 6-inch $\times$ 12-inch stock journal bearing. Tests also were made under normal operations of a 30 $\times$ 60-inch bearing which was part of a 60,000-k.v.a. generator.

Bearing performance can be calculated from the characteristic number $\frac{ZN}{P}$, where Z equals viscosity in centipoises (depending upon temperature and grade of oil), N speed in revolutions per minute, and P unit pressure of projected area in pounds per square inch. Coefficient of friction and minimum oil film thickness of a journal bearing are shown to be nearly independent of variations of unit clearance between rather wide limits during which the bearing operates near the best condition of minimum coefficient of friction and maximum oil-film thickness. The calculation of bearing performance is thus made easy.

LARGE BEARINGS.

An accurate experimental investigation of sleeve-bearing oil films on a large-size power bearing was described in "Film Lubrication in Sleeve Bearings," by M. Stone. Results of this investigation show a sufficiently close agreement of theory and practice to allow designers to make minimum oil-film thickness a reliable criterion of bearing performance. This can be considered as a refutation of Wolff's results, using an interferometric method, subject to definite objects.

OIL-FILM BEARINGS.

"Current Practice in Pressure, Speed, Clearances, and Lubrication of Oil Film Bearings," by H. A. S. Howarth, contains much design and operating data furnished by makers of bearings for the trade and for machinery of their own manufacture. Most of the data are tabulated and classified on charts, the most useful of which has as its coordinates pressure and velocity. Upon this chart are applied special scales showing some of Dr. Albert Kingsbury's "Optimum Conditions for Journal Bearings." Examples show how analytical laws are useful in solving simple, practical problems of bearing design.

Selection of a lubricant of suitable viscosity is explained by a chart that may be entered with calculated absolute-viscosity coefficient at operating temperature, and will yield the required commercial viscosity in Saybolt or Redwood seconds or Engler number for a mineral oil of any base.

DISCUSSION.

Experiences related when presenting and discussing these papers clearly indicated that the art of designing and lubricating journal bearings has been developed until theory and practice agree quite closely. Performance of bearings and their lubrication can now be predicted with a high degree of certainty. If bearings are operated under conditions for which they are designed, long life and satisfactory service can be expected.

It was pointed out that the load carried by a bearing depended upon the ratio of its length to its width. Loads of over 10,000 pounds per square inch have been carried on journal bearings before failure of the oil film and bearing.

Operation of bearings at slow journal speeds was the subject of considerable attention. From this discussion it was evident that some of the fear of heavily loaded high-speed bearings failing when operating at slow speed, such as when stopping and starting, is not well founded. Some of these bearings have been operated at 1 R.P.M. without injury. Even if only a partial oil film is maintained, the journal is moving so slowly that if metal-to-metal contact should occur sufficient heat would not be produced to damage the bearing. Kingsbury thrust bearings will operate at speeds as low as 5 to 1 R.P.M. before there is any indication of oil-film failure. In one case a bearing having a mean diameter of 9 inches was operated heavily loaded at speed of less than 1 R.P.M. for a considerable period.

Tests have shown that the oil-film thickness between a journal, running at normal speed, and its bearing varies through a cycle so that a point on the end of the shaft describes a closed figure. Electrical gauges have been developed that are mounted on the bearing and make possible measuring shaft movement of one-millionth of an inch.

The necessity of maintaining good alignment of bearings was emphasized. If bearings are not in good alignment with their journals, in some part of the bearing the oil film may be thick and in others it will be very thin or even ruptured. Bearings may be properly aligned on the bedplate before the rotor is in place, but because of bearing pedestals and bedplate distortion, due to the rotor weight or to other causes, the bearings may be out of line after the machine is assembled.

Tests made on a Kingsbury type thrust bearing showed that higher loading pressures could be carried on babbit metal than on bronze. In one case where a bearing of this type was loaded to destruction, bronze failed at loads of about 14,000 pounds per square inch and babbit-faced shoes carries over 16,000 pounds per square inch before failure.

In answer to a proposal that when laboratory tests were made care should be taken to duplicate practical conditions in oil and other factors, it was suggested that laboratory conditions be duplicated in practice. Bearings can be designed to carry heavy unit loads, operate efficiently and have long life. To do this, however, they must be supplied with clean oil of the proper grade and advantage taken of means available for keeping the lubricant in good condition. Laboratory conditions must go into operation of bearings if they are to operate successfully under loads carried in the laboratory.—"Power," Mid-December, 1934.

BOILER FEEDWATER.

Like that of medicine the science of boiler feedwater treatment and conditioning is important and endless. In the search for dollar-saving water facts many thousands of dollars and untold time have been expended on researches under the auspices of the Boiler Feedwater Studies Committees, of which

Sheppard T. Powell is chairman. Results previously reported have been well worth the cost in reduced outage and maintenance and increased efficiency of powerplant boilers. Additional progress was revealed in the two feedwater sessions this year.

NAVY LIKES 1933 COMPOUND.

"Power" for Mid-December, 1933 (page 687), announced the U. S. Navy's tentative standardization on a new boiler compound ("Navy Compound 1933"). Its formula was selected after extensive operating tests costing thousands of dollars. The constituents of this compound are:

Anhydrous disodium phosphate	47 per cent
Soda Ash	44 per cent
Corn starch	9 per cent

This has now been given an extensive trial in fifty ships of the fleet. Results as presented by Dr. Robert C. Adams, Jr., chemical engineer, U. S. Naval Experiment Station, Annapolis, Md., are distinctly favorable. Over two-thirds of the ships reported no clouding of gauge glasses. Practically all said there was no tendency to prime. Most of the ships reported a decrease in corrosion and only one an increase. Commenting on his paper ("Results from the Change in the Navy's Method of Boiler Water Treatment") Dr. Adams again warned against its indiscriminate application to land plants or merchant marine. On Navy ships, the character of impurities is fairly constant and notably different from that of most land plants.

DOES SULPHITE KILL OXYGEN?

For years dissolved oxygen has been recognized as a primary cause of corrosion. Today much money is spent to get it out of feedwater, and keep it out. To get it out, mechanical deaeration has generally been used. Special deaerators or deaerating heaters "boil off" the air and pass it off with the vented vapor. But, it is well known that certain chemicals combine with dissolved oxygen. Some of these, notably sodium sulphite, have been considerably used for this purpose in the past year or two.

DISCUSSION.

Dr. R. E. Hall (Hall Labs., Inc.) reported several years of operation of the boiler plant of the Pittsburgh Steel Co., Monessen, Pa. Monongahela River water is dosed with a special sulphite mixture and then fed directly to 400-pound boilers with no evidences of corrosion.

The importance of getting the correct grade of sulphite was stressed by several speakers. Some suggested using proprietary brands to insure correct quality. Others strongly urged the preparation of complete specifications that could be handed to purchasing agents for competitive price buying.

One speaker said that purchasing agents could get engineers what they need if engineers will be more specific in requests.

S. P. Applebaum, Permutit Corp., referred to cases of economizer corrosion where sulphite was used and suggested that inserting a tank in the line to give ample reaction time might eliminate this trouble. There was no corrosion in the boiler concerned.

E. H. Barry, power supervisor of West Virginia Pulp & Paper Co., said he had gotten excellent deaeration with ordinary open heaters vented liberally. The resulting water had been satisfactory for all boiler pressures from 45 to

600 pounds. Certain cases of economizer corrosion led him to suspect that stagnant water (due to poor circulation) is much more likely to cause corrosion.

Mr. Powell warned against attempts to use sulphite as the sole means of removing oxygen. Mechanical deaeration should always come first, he said, leaving sulphite, if used, to remove the last traces only.

Another discussion developed on the use of tannin for boiler dosing. Many comments were favorable. Several speakers reported use of tannin for long periods without tube failures or traces of corrosion. Captain Dinger reported experiences with a number of marine boilers where tube loss had stopped as soon as tannin treatment was started.

WAR ON EMBRITTLEMENT.

In another feedwater session was presented "A Critical Summary of Published Information Relating to the Embrittlement of Boiler Steel." This is part of a survey made in cooperation with the Bureau of Mines. Results have not yet been released for publication.

Two other research papers of specialized interest were "The Solubility of Sodium Sulphate in Boiler Water Salines As Related to the Prevention of Embrittlement," and "The Effect of Solution Concentration on the Failure of Highly Stressed Steel," both reports by W. C. Schroeder and E. P. Partridge.—"Power," Mid-December, 1934.

WELDING *VS.* CASTING.

For the last few years, proponents of that lusty young infant, welding, have been itching for a real argument on welding *vs.* casting. Their success against riveting and bolting has been notable, but somnolent foundrymen have suddenly awakened and gone to work to better their products. The result is better castings to compete or cooperate with better welding, in any case giving the power engineer stronger and better products capable of standing higher stresses, higher pressures, higher temperatures.

STEEL VS. CAST IRON.

An extensive comparison of properties was presented in "Steel and Cast Iron in Stress-Transmitting Mechanisms," by Everett Chapman. General conclusions were that commercial gray iron is an aggregate of pure graphite in flake form and impure steel, the graphite flakes supporting the matrix in compression but starting incipient cracks in tension or torsion. Welded steel, on the contrary, is 2½ times stiffer and excellent in tension and torsion as well as compression, providing *welds are properly made.* Cast iron "grows" much more than steel under repeated heating and cooling.

Modern weld metal is a freak. It is perfectly possible to produce in the electric arc, in the form of weld metal, a steel with carbon content of 0.08 per cent, tensile strength of 65,000 pounds per square inch, and elongation 35 per cent in 2 inches, much better than anything producible by the open-hearth process. But, to carry the analogy further, the latent tensile strength of open-hearth steel is about 500,000 pounds per square inch, while the usual values in practice are about a tenth of that figure. In other words, man has not yet learned to process steel to obtain its maximum properties.

Among Mr. Chapman's illustrations of newer power units designed for steel is the engine frame illustrated. This is completely welded up from plates and combines light weight with the ability to sustain high impact loading for long periods of time. The gas load of 38,000 pounds is applied at the bolt circle holding the cylinder head to the top deck, and the reaction to the gas

load occurs at the large studs that secure the main bearing cap to the frame. V-type construction further complicates the loading, since a joint must be put in between two elements in tension.

Cromansil steel with flame-cut contours provides the necessary strength and avoids sharp (and weak) corners. Machining also was used in order to produce a generous curve at the point of intersection of the top deck and the side sheets. It is sound practice in a structure of this character definitely to control these critical sections, and welding in general cannot be relied upon to produce a predictable contour in a corner. A careful technique must be specified and supervised in making the joints between the component parts in order further to insure the absence of uncontrolled boundaries, either internal or external. Once this procedure has been followed, the only limit to the lowering of the factor of safety is the residual uncertainty as to how far this process has been carried, not only in the construction of the piece but also in the actual selection of the steel entering into the structure. In the ultimate, then, the strength of a structure could increase indefinitely in proportion to the cleanliness of the materials and cleanliness of the design.

CASTING VS. WELDING—BY A USER.

S. Einstein, chief engineer of Cincinnati Milling Machine Co. and Cincinnati Grinders, Inc., discussed the subject from the user's point of view. Whether a cast or welded structure is to be used for machines or machine parts depends on many factors. The average machine shop has very little or no experience in welding. The majority of designers, if asked to design a welded structure, incorporate in it the experiences and ideas which they had on cast iron. Thus advantages of the welded structure are greatly minimized.

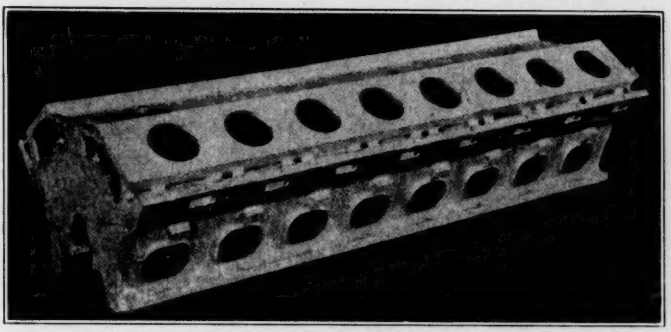
Welded structures in general are lighter in weight, have greater strength proportional to their weight, enable quicker delivery, and result frequently in more stable structures. Theoretically, the cost of such parts, if only made singly, is cheaper; and under certain conditions the cost of parts manufactured in very large quantities with elaborate tooling might be cheaper. Cost of patterns is eliminated for welded structures, but the necessity for templates or other auxiliary tools reduces this saving.

The disadvantage of welded structures is the greater possibility of vibrations—the cast structure dampens vibration, while the welded structure may resonate. The welded structure on a medium-size production job results in higher cost, and this is particularly so when cast iron or hardened steel ways have to be attached to such a structure for sliding surfaces. To reduce internal strains, welded structures quite often have to be annealed, a further item of expense. The welded structure does not lend itself so readily to production of pleasing lines and contours. Generally speaking, the best and cheapest welded structure is the structure which has a minimum of welded joints.

Mr. Einstein's company normally welds designs of which only one unit is required, considers the merits of both casting and welding when 3 to 5 units are required, and above that number uses castings.

DISCUSSION.

Mr. Horgan of General Electric brought out in discussion that the rapid ascendance of welding has put the foundryman on his mettle, after years of slumbering security, the result being better products for the user. Other discussers brought out present high costs of welded construction, the tendency toward greater vibration with steel, the fact that steel castings and rolled steel are now being combined by welding.—" Power," Mid-December, 1934.



V-Type Diesel Frame, Welded of Cromansil Steel.

WASTED EFFORT IN THE PACIFIC.

At the moment of writing there sems to be no certainty about whether Mr. C. T. Ulm and his companions on the flight from California to Hawaii are alive or dead. In view of the importance popularly attached to the phrase *De mortuis nil nisi bonum* we may assume that they are alive and are waiting on some eligible atoll, with what the house agents call "vacant occupation," to be picked up—much as Messrs. Ulm and Kingsford-Smith, not then knighted, were missing for some time in the Australian bush while two very good men died in the process of looking for them.

The whole affair raises again all the points that we have ever put forward in this paper against the folly of those trans-oceanic flights in unsuitable aeroplanes. The only use of such flights is that they emphasize the fact that there is no suitable aeroplane. The Airspeed Envoy was just as good, or just as bad, whichever way you like to look at it, as any other aeroplane for the job. The last radio messages received indicated that the machine had run out of petrol, so obviously if the machine did come down in the water no blame can be attached either to the aeroplane or its motors.

This makes all the more disgusting the messages from Oakland, California, in which American mechanics are reported to "have disclosed that they privately expressed the opinion that the plane was unsafe, declaring that the fabric was patchy and that loose ends were hanging from the wings and the fuselage." One imagines that the "mechanics" who were responsible for this libel were merely hangers-on of some aerodrome, who saw the machine in its wrappings before it was unpacked, or perhaps saw it standing in a shed with some cleaning cloths hanging about it. Certainly Airspeed, Ltd., are not accustomed to sending out machines with patched fabric and loose ends hanging about. To call it a "suicide crate" was justifiable, for any aeroplane which set out on such an adventure might be so called equally fairly. The only exception might be a properly organized flight by units of the U. S. Navy.

There is some interest in another message from California which says that Air Commodore Sir Charles Kingsford-Smith said that "he would himself take off to help in the search, but that his ship was in the factory undergoing repairs." The last we heard of Sir Charles was that his ship had the bailiffs in for an alleged debt incurred during his previous visit to California.

Sir Charles Kingsford-Smith had only just taken an equally foolish risk in doing the same trip in the opposite direction without any understandable reason, except his need for publicity. And we have too much respect for his good sense to imagine that he was willing to take off from California in a single-motored ship to fly some 3000 miles across the ocean to Hawaii to look for Ulm, when according to other reports the whole of the U. S. Naval Air Service and Army Air Corps machines in the Hawaiian Islands had already been looking for Ulm for days, and everything in the way of ocean traffic within 500 miles of the islands, all the way from luxury liners down to what a report from Honolulu calls Japanese sampans were searching for him. Incidentally, a sampan is a Chinese vessel and not a Japanese, but we must allow some journalistic license in such reports.

According to another report from Honolulu twenty-four aeroplanes belonging to the U. S. Navy and ten belonging to the U. S. Army had been searching the ocean for Ulm and his misguided companions. Even allowing that perhaps a dozen of these may have been flying-boats and would stand a chance of surviving if they came down in calm water, that means that anything between 70 and 100 expensively-trained and therefore valuable aviators have been risking their lives, and the money which they cost to train and their machines, in looking for people who started out deliberately on a useless voyage, purely for the sake of publicity.

The voyage was advertised as a survey of the route to Australia. Actually it was a stunt. Years ago we defined a stunt as something that is difficult and/or dangerous to do, and not worth doing when done. And surveying the Pacific from an aeroplane only reminds one of the War-time "Punch" picture of a Coastguard stopping a painter from painting a sea-and-cloud-scape because the picture "might get into the hands of the Germans."

If Ulm and his companions had got to Hawaii they would have proved nothing, because the trip has been done many times over. It was done first in 1927. And an idiotic race to Hawaii cost a number of lives because so many competitors fell in the sea or were killed at the start through trying to get off with overloaded machines. It has been done since by the U. S. Navy's flying-boats as a regular Service voyage with proper radio arrangements and proper patrols of warships,—which is the only sensible way. And it was done by Ulm himself in 1928, when he accompanied Charles Kingsford-Smith with a wireless operator and a mechanic in a Fokker tri-motor monoplane, the old *Southern Cross*, in a flight from San Francisco to Sydney. Consequently, there was no need for anybody to do it again in a land-plane.

Besides risking the lives and property of the U. S. Navy and Army this ridiculous affair has wasted the time of Admiral Yarnell, Senior U. S. Naval Officer in Honolulu, and even of Mr. Swanson, the Secretary of the U. S. Navy at Washington, and it has deflected from their courses a number of quite important ocean-going steamers, which consequently have used more coal and wasted so many man-hours in doing so, apart from the fact that passengers who were in a hurry will have arrived late in consequence.

Years ago we said in this paper that the Governments of the World should announce publicly that no official search will be made for any aviators who go amissing on foolish voyages of this sort. And we suggested that the shipping companies likewise should issue orders to their skippers that they were not to deflect from their courses to search for such people.

The French Government, with that ruthless feminine logic which is one of the best characteristics of the Mediterranean French, long ago announced that nobody would be allowed to fly across the Sahara unless they deposited enough money to pay for a search being made for them by the *Armée de l'Air* if they did not arrive at their destination by a certain fixed time. This has naturally stopped several honest adventurers from flying across the Sahara. But probably it has stopped just as many publicity stunters who might have camped in some oasis till the whole French African Air Force had been hunting for them for days and they had become the talk of the moment in the Excitable Press.

Nations cannot very well exact a cash deposit from trans-oceanic fliers, but they might stop a certain number of them, and they would certainly give the public the proper aeronautical outlook on such performances, if they were to announce that no search whatever will be made in future for missing trans-oceanic fliers.—"The Aeroplane," December 12, 1934.

DIESEL PROGRESS.

Much hope for the future of the Diesel engine was given in the combined Power Progress Report (replacing individual Division reports for the first time this year) by C. F. Hirshfeld, chief of research, Detroit Edison Co. While bad economic conditions have meant fewer power projects generally, the Diesel industry has developed in "almost spectacular style." This is due to increased rotative speeds (giving lighter or more powerful engines), increasing use of superchargers, improved design and materials (meaning better performance and less maintenance), and to solid injection (meaning

simplified installations). Greatest fields today for the Diesel are in transportation and in the small or comparatively small stationary plant, where the greater simplicity and higher thermal efficiency of the Diesel shows to greatest advantage. Utility companies are beginning to see the merits of the Diesel for small isolated plants to avoid long, expensive transmission lines. But Dr. Hirshfeld injected the note of warning that a future important competitor of the Diesel will be the small, single-unit, high-pressure and high-temperature steam plant. Though its thermal efficiency is lower, its economic efficiency (the efficiency which *sells* prime movers) may be higher due to cheaper fuel.

Dr. Hirshfeld also mentioned use of alloy cast iron for crankshafts in automotive engines and high-speed Diesels.

MOTORSHIP'S FUTURE.

Louis R. Ford, in "Future of the Motorship in the United States," outlines an anomalous situation in which the United States is found to be the only maritime nation that has thus far clung to steam machinery exclusively for propulsion of its large, seagoing merchant ships. So extensively has the Diesel engine been applied to all forms of marine propulsion in foreign countries that now 58.3 per cent of all tonnage building throughout the world is equipped with Diesel machinery. Of the total motorship tonnage under construction throughout the world the United States is building but 0.9 of 1 per cent.

The imminence of a cargo ship building program in the United States, as forecast by President Roosevelt's positive statement that subsidy legislation will be enacted by the next Congress, suggests the advisability of an analysis of the situation, with a view to determining what are the factors that have influenced American shipowners to prefer steam to Diesel machinery.

DISCUSSION.

Mr. Huckle brought out that the new Thomas-Richotte system of producing full-automatic Diesel sets using several small engines (described recently in "Power" by Mr. Thomas) is now on trial in an installation for a Nevada "dude" ranch, and that another installation, using three Ford V-8 engines, is soon to be tested. If successful, an installation of three Diesels will be tested on a fishing boat.

FUEL STANDARDIZATION.

Dr. P. H. Schweitzer, of Penn State College, led discussion on this point, reporting failure of the combined A.S.M.E.-S.A.E. Fuel Oil Standardization Committee, and announcing the recently proposed A.S.T.M. tentative standard. He advocated conciliation and cooperation in settlement of this important problem.

DISCUSSION.

Several discussers brought out that they were at present using fuels that fall between two of the grades shown in the A.S.T.M. tentative standards, and that in certain cases the gaps are too wide between grades. It was the opinion of discussers that the grades should be modified, possibly even combined with grades of domestic furnace oil to decrease tankage and simplify supply. O. F. Allen and Mr. Pender, of Sinclair, were among those expressing this view.—"Power," Mid-December, 1934.

AERONAUTICAL ENGINEERING.

The Case for the Composite Aeroplane.

The "Composite Aeroplane" of which the prototype is now being built by Short Brothers (Rochester and Bedford) Ltd. was briefly referred to in "The Aeroplane" last week. The premature and unauthorized story in a daily paper which brought about that statement has drawn attention to the scheme earlier than its sponsors, which include the British Air Ministry and Imperial Airways Ltd., had desired.

For these reasons Major Mayo does not yet feel free to write for "The Aeroplane" the article about his invention which he promised to give us at the appropriate time, but what follows is believed to set out the considerations which have led to the building, after the most exhaustive preliminary examination of the project from all aspects by such authorities as the Air Ministry and Imperial Airways Ltd., of the first Mayo Composite Aeroplane.

THE PURPOSE OF THE INVENTION.

The Mayo Composite Aeroplane is admittedly an invention of a revolutionary kind, but its object is to provide a practical solution to the fundamental problem which has hampered the development of long-range flying ever since the Wright Brothers produced the first successful aeroplane. During these thirty years of intensive development designers all over the world have recognized that the performance and effectiveness of aeroplanes could be very greatly improved by designing for much higher stalling speeds, but it has hitherto been impossible to take full advantage of this potential improvement in performance owing to the difficulties and dangers of taking off and landing machines which have very high stalling speeds.

Such extreme examples as the Schneider Trophy seaplanes have clearly demonstrated that a machine which has a very high stalling speed can fly quite safely and highly efficiently when once it has been got into the air.

The main purpose of this invention is to make possible the full realization of this great potential advance in performance by providing means by which an aeroplane of extremely high loading can be taken into the air and launched with complete safety. The practicability of the scheme depends, of course, on making provision for the safe landing as well as the safe launching of such a machine. At the end of its flight the aeroplane must, in fact, have had its stalling speed so far reduced as to ensure a normal and safe landing.

This imposes the condition that a considerable part of the load carried must be of a consumable or dischargeable nature. This is the condition which defines the field of application of the invention, whether in commercial or in military aviation. In any long-range aeroplane the load must consist largely of fuel and oil, and this load must necessarily have been consumed by the end of the journey. The load of a military aeroplane may also be discharged in the form of bombs. In either instance the machine will arrive at its destination with its total weight greatly reduced, and in this condition it will have a normal landing speed.

In the rare event of having to make an emergency landing before completion of the journey, the pilot will be able to reduce the landing speed to the safe limit by jettisoning fuel or other load. The elimination of the fundamental objection to a high stalling speed at the start opens the way to great advances in speed, range and load-carrying capacity and corresponding reductions in capital and operating costs for long-range work. The following is a summary description of the invention:

MAIN PRINCIPLES.

The aeroplane to be launched is mounted rigidly on top of another aeroplane to form what the inventor has called a "Composite Aircraft." The lower component is given characteristics such that the composite aeroplane as a whole will have a moderately low stalling speed and ample power to ensure a good take-off in any conditions. Each component is manned by the crew needed to fly it separately.

The composite aeroplane is thus able to take off quickly and easily as a single unit from land or water, as the case may be, as the two components are securely locked together during this process. When the composite aeroplane has climbed to a safe height it then accelerates until it attains a speed considerably above the stalling speed of the upper component, even though that stalling speed may be far higher than the standard of today.

The key to the practicability of the invention is the provision of means by which, when the necessary flying speed has been attained, the upper component can be detached from the lower component and launched separately without any possible risk of fouling or failure.

To achieve this, two main alternative or complementary methods are provided under the several patents. One depends on a relative increase in angle of incidence of the upper component during the flight of the composite aeroplane, and the other on the selection of wing sections which automatically ensure a transference of lift from the lower to the upper component as the composite aeroplane accelerates.

By one or other of these methods, or a combination of both, a substantial separating force of predetermined magnitude is established before the upper component is allowed to cast loose. A number of important safety devices are incorporated, and these, in conjunction with the prevention of detachment except when rapid vertical separation must result, ensure continued and controlled flight of the individual components after their separation, without risk of fouling or collision between them.

After separating, the upper component proceeds on its way to perform its intended function and the lower component returns to its base and becomes available for the launching of another upper component.

The whole process of starting the aeroplane to be launched on its journey need take only a few minutes, so that one lower component can, if necessary, launch successively a number of upper components in quick succession. On the other hand, if the upper component is designed specially for high flying, the separation may be delayed until any desired height has been reached.

The practical operation of mounting the upper component on the lower presents no serious difficulty, and the cost involved is comparatively unimportant in relation to the advantages gained. The lower component can be used alternatively as a normal commercial aeroplane, if desired.

The aeroplane to be launched would generally be a high-performance machine, with small wing-area and correspondingly high stalling speed when loaded. The lower component would have relatively large wing-area and would make up for any deficiency of the upper component in this respect. But it can combine this feature with a relatively low weight-power ratio, for the reason that it is not required to carry any useful load beyond fuel for a few minutes' flying.

The combination of the two components can thus be given an exceptionally good take-off, a feature of obvious importance, particularly for sea-going work.

If the aeroplane to be launched need not have a very high performance, it may have a high initial stalling speed in combination with a comparatively higher weight-power ratio, and the lower component may be arranged to provide whatever power is needed to ensure a good take-off for the composite aeroplane as a whole.

The invention can, in fact, be applied to the launching of any type of aeroplane whose characteristics are such as to make it difficult to take off. It can, for instance, be applied to the launching of an aeroplane designed for extreme ranges.

According to present practice, land-planes can be designed for ranges of between 5000 and 6000 miles, but such machines are impracticable because they can only be taken off in exceptional conditions. Its inventor hopes that the composite aeroplane will not only make such machines practicable for ordinary military or commercial operations, but will also allow a considerable increase in the cruising speed for which they can be designed.

A further technical advantage should be mentioned. The variable-pitch airscrew is an almost essential corollary of high wing-loading in normal practice, but it involves substantial additional cost and weight, particularly if the advantages of supercharging are to be got. The composite aeroplane not only allows far higher wing-loadings than have been contemplated so far, but does so without needing a variable-pitch airscrew at all, even when full high-level supercharging is used.

The reason for this important technical gain is that the heavily loaded upper component is relieved of the burden of having to take off by its own power.

APPLICATION.

In commercial aviation the main application of the invention will naturally be on long-range services. Commercial loads are not generally dischargeable, though mails or other load may be dropped by parachutes in the future. This invention will make ranges possible in commercial aviation which have hitherto been impossible. Its most obvious application is to the direct North Atlantic service.

The successful flights across the Atlantic have only been made in the most favorable conditions, and often only after weeks of waiting for the right conditions. And even then the failures have been almost as numerous as the successes.

The fundamental problem has been to obtain enough range at high enough speed to make the crossing sure in bad weather, and this has stopped even an attempt at a commercial service. The composite aeroplane will, it is hoped, give enough margin of range for a service to be maintained through the worst Winter weather, when continuous head winds of 50 M.P.H. and even more may have to be faced.

Owing to the relatively small size of aeroplane needed for a given pay-load, the cost should be far less than that of another trans-Atlantic scheme which has recently been much discussed.

For military aviation, the application is obviously not confined to long-range work.

Incidentally, the composite aeroplane can be made to take off so easily that it should be able to operate at night quite safely—which is very important in both military and commercial aviation.

Many people have said that only the largest flying-boat would be seaworthy enough to face Atlantic weather, but such statements display ignorance of facts. No aeroplane of a size within present practical politics could live on a real Atlantic sea for more than a few minutes. The only way to ensure safety and reliability is to stop forced landings.

The aeroplane must be multi-motored and able to fly on if one motor stops. And it must have enough range to get it across in the worst weather. We hope that the composite aeroplane will combine in its relatively small upper component these features, which are essential in an Atlantic Service.

COMPARISONS WITH A CATAPULT.

In reviewing the general scheme of the Mayo composite aeroplane we see that the lower component performs in a sense the functions of a powerful and mobile catapult. But it does something which is beyond the scope of a catapult, namely, to launch the heavily-loaded service aeroplane at a safe height.

Let us remember that a service aeroplane could not be expected to bring off a safe landing in its fully loaded condition. Even if it could be got into the air by means of a very powerful and lengthy catapult, it would be launched just above the surface of the ground or water and at a speed not far above its stalling speed, and would thus be in a potentially dangerous condition. The failure of a motor might force a landing before the pilot had time to jettison the fuel.

But apart from the question of safety, a catapult of the size and power needed would involve considerably higher cost and would lack the advantage of mobility which can be claimed for the composite aeroplane scheme.

The possibilities of refuelling in the air as a means of achieving very long range should also be mentioned. The technical difficulties involved in refuelling are serious enough in clear and steady weather, but in even moderately bad conditions these difficulties become much greater. In such weather as that which must be faced for a regular Atlantic service refuelling in the air would be quite impracticable.

The press has suggested that the composite aeroplane will be suitable only for mail services, but this is not necessarily so. The development of Atlantic services must go in stages, and the first stage must clearly be a mail service. For this reason the Mayo composite aeroplane is being developed in the first place with the Atlantic mail service in view, but there seems to be no fundamental reason why pasenger-carrying types should not follow at a later stage. Whether the load is to be mail or passengers the really vital requirements for an Atlantic service—high speed in combination with reliability and an ample margin of range—are the same, and to meet these essential requirements the composite aeroplane scheme has been evolved.—"The Aeroplane," December 5, 1934.

A NEW NON-FERROUS ALLOY.

Two Air Ministry specifications, D.T.D. 232 (hard), and D.T.D. 237 (soft), have just been published for a new non-ferrous alloy of nickel, copper and zinc, which has been developed by William Gallimore and Sons, Ltd., of Sheffield. For certain applications the alloy seems likely to be a serious competitor of stainless steel. Extensive tests in sea water have shown that there is no corrosion in the metal itself, or at joints made with grade "A" tinman's solder. The physical properties of the alloy are comparable with those of materials to specifications D.T.D. 166A and D.T.D. 171A.

In the fully annealed condition (D.T.D. 237) the 0.1 per cent proof stress is in the neighborhood of 16 to 18 tons per square inch and the maximum stress about 40 tons. In the hard rolled condition (D.T.D. 232) the 0.1 per cent proof stress is about 55 tons per square inch.

The new alloy can be readily machined or cut. It will bend and take shapes with ease, though intermediate annealing may be necessary if such manipulation is complicated. The annealing temperature is about 780 degrees C. The alloy can be air-cooled or quenched in water.

Annealing does not cause scaling or have any other effect on the surface except to dull slightly the polish. The alloy can be easily soldered, and strong joints can be made with the usual grade "A" Tinman's solder. The electrolytic action set up is negligible.

When the alloy is in juxtaposition to the non-corrodible austenitic steels or the lighter alloys such as Duralumin, the electrolytic action is so slight as to be of no consequence. The alloy is not liable to fatigue or season cracking. It can be readily polished. It is white in color, and undistinguishable in appearance from steel. The scrap is of value according to the prices in the scrap market for the component metals.—"The Aeroplane," December 5, 1934.

NORTH ATLANTIC CATAPULTS.

Since the Cunard White Star ship *Queen Mary* was launched on September 26, one has heard the question whether any arrangement has been made to equip the ship with catapults for shooting off mail-carrying aeroplanes before reaching port. The success of the catapult system operated by the Deutsche Luft Hansa in connection with the Nord Deutscher Lloyd ships, *Bremen* and *Europa*, prompts this inquiry.

Mr. Simmonds asked the Chancellor of the Exchequer in the House of Commons on November 14 whether the owners of the *Queen Mary* were under any obligation to H. M. Government in regard to catapults for aircraft, and Mr. Cooper replied that they were not.

"The Aeroplane" took this matter up with the Cunard White Star, Ltd., some weeks ago, and the company stated that it had considered the proposition, and, as a result of careful inquiry into the use made of this facility in other ships, and after a consultation with the postal authorities, the company decided not to install the necessary equipment in the *Queen Mary*.

On October 29 "The Daily Mail" stated that the present plans of Cunard White Star, Ltd., were understood to include provision for the carrying of two air liners for passengers and two aeroplanes for mails, and that these would probably be able to take off from the decks of the *Queen Mary* six or seven hours before she is due in port. We again approached the Cunard White Star Company, and they replied that this statement was entirely unauthorized and without foundation.—"The Aeroplane," December 5, 1934.

REAR ADMIRAL JOHN HALLIGAN, U. S. NAVY

PAST PRESIDENT OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS

REAR ADMIRAL JOHN HALLIGAN.

It is with the deepest regret that we have to announce the death of one of our most distinguished members, a past President and also a past Secretary-Treasurer of this Society. Rear Admiral John Halligan died at the Naval Hospital, Puget Sound, Washington, on 11 December, 1934.

Admiral Halligan was born at South Boston, Massachusetts, on May 4, 1874, and was appointed a Naval Cadet from that State in 1894, standing at the head of his class upon graduation in 1898. His career in the Naval Service was varied and he excelled in every department of that service with which he was connected. However, it is his close connection with and his many contributions to the engineering branch of the Service which will be remembered most by the members of this Society.

Admiral Halligan was among the first officers in the Navy connected with the investigation, experimentation and development of the use of liquid fuel. As a Lieutenant, he served as Commanding Officer of the Torpedo Boat *Rodgers,* one of the two vessels used to conduct extended experiments in the fuel oil burning tests authorized in 1902 under the auspices of the Liquid Fuel Board. Perhaps no greater tribute could be given to any naval officer than that contained in the Board's official report and quoted as follows:

" The Commanding Officer of the U. S. Torpedo Boat *Rodgers,* Lieutenant John Halligan, Jr., U. S. N., showed in himself such an example of zeal, efficiency, and professional ability that his personal work and service was, to a considerable extent, the inspiration that caused the enlisted force of the Torpedo Boat to secure results that have in many respects never been surpassed in experimental research and investigation.

" The Board has no hesitation in stating that, if it had not been for the untiring, intelligent, executive work of the Commanding Officer of the U. S. Torpedo Boat *Rodgers,* no such extended series of tests could have been conducted during the period that the

Board was engaged in this special work. As typical of his zeal and engineering ability, as well as of his physical and moral courage, it is only necessary to state that in every time of possible danger he was always among the first to open and close the necessary valves to restore normal conditions. * * * The tact and judgment displayed by Lieutenant Halligan in his dealings with numerous inventors was likewise a determining element in having all recognize the fact that, while the personal sympathy of the Naval Officers was with each and every inventor, such sympathy would not stand in the way of rigid consistency and integrity in collecting absolutely correct data and information.

"The earnest purpose of this officer to acquaint himself with the manner of conducting experimental investigation and research, and his high administrative efficiency in stationing and directing his crew so that the results of the experiments would commend themselves to all interested in the solution of the oil-fuel problem, is worthy of special recognition by the Department, and the Board requests that this deserved tribute to his professional zeal and engineering ability be made a part of his official record."

Throughout his subsequent career, the qualities exemplified by Admiral Halligan in the foregoing quotation characterized all his activities, and have been and will be a constant source of inspiration to members of the engineering profession everywhere. In no less distinguished a manner were all his duties performed, including those for several years as Chief of the Bureau of Engineering, to which office he was appointed in June, 1925.

With his passing all engineers have lost an inspiring leader and a true, kind, loyal and sympathetic friend.

BOOK REVIEW.

LANGE'S HANDBOOK OF CHEMISTRY.—By Norbert Adolph Lange, Ph.D., Handbook Publishers, Inc., Sandusky, Ohio.

This handbook covers a wide scope in the field of chemistry. For handy reference, the chemical and physical properties are given principally in tabular form, thus allowing the presentation of the maximum of information in the minimum of space.

In addition to the chemical information, there is also a great deal of information ordinarily found in engineering handbooks. Consequently, it will be found that although this book was written primarily for the industrial chemist, it will prove very useful to the student chemist, metallurgists, physicists, engineers and many others.

ASSOCIATION NOTES.

ELECTION OF OFFICERS FOR 1935.

The following have been elected as officers of the Society for 1935:

President:

Admiral Wm. H. Standley, U. S. Navy.

Secretary-Treasurer:

Commander C. S. Gillette, U. S. Navy.

Members of Council:

Captain N. H. Wright, U. S. Navy.
Captain B. H. Bruce, U. S. Navy.
Captain Henry Williams (CC), U. S. Navy.
Commander S. S. Kennedy, U. S. Navy.
Commander H. N. Perham, U. S. Coast Guard.
Mr. William M. Corse.
Mr. John F. Nichols.

ANNUAL DINNER.

The Annual Dinner of the Society will be held on Friday, March 29, 1935, at 7:30 P. M., at the Willard Hotel, Washington, D. C. Unusual interest is being shown in this event this year and it is believed that it will be outstanding in a long series of very successful occasions of this kind.

MEMBERSHIP.

The following members have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Augustine, L., Lieut. Commander, U. S. N. R., Barnegat Pines, Forked River, N. J.

Christy, Robert E., 1100 Bay St., Alameda, Calif.

Gurley, Ralph R., Lieutenant, U. S. N.

Kane, Lail, Lieutenant, U. S. N. R., 143 North Hope St., Los Angeles, Calif.

Perkins, Sidney Davis, 15 Buckthorne Ave., Providence, R. I.

Porter, Kenneth, Lieutenant, U. S. N. R., care Inspector Machinery, U. S. N., Camden, N. J.

Quigley, Francis P., Lieut. Commander, U. S. N., Federal Power Commission, Washington, D. C.

CIVIL.

Davies, James A., 317 North Chester Road, Swarthmore, Pa.

Mildon, Reginald B., Vice President, Westinghouse Electric and Manufacturing Co., Lansdowne, Pa.

ASSOCIATE.

Campion, Edward W., General Manager, The Bonney-Floyd Co., Columbus, Ohio.

Parker, J. Irwin, 1629 Leighton Ave., Los Angeles, Calif.